# CHARACTERISTIC OF DRIFT ELIMINATORS OF AN EVAPORATIVE CONDENSER WITH FORCED AND INDUCED FLOW

*by*

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**AMIT KUMAR SINGH**

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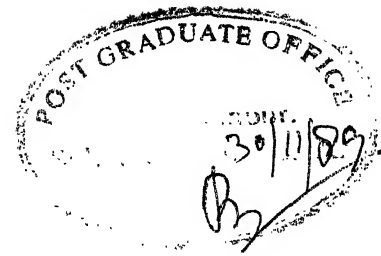**INDIAN INSTITUTE OF TECHNOLOGY KANPUR**

**NOVEMBER, 1989**

ME-1589-M - SIN-CHA

TO MY

PARENTS, BROTHERS AND SISTERS

## CERTIFICATE

This is to certify that the work entitled
'CHARACTERISTICS OF DRIFT ELIMINATORS OF AN EVAPORATIVE
CONDENSER WITH FORCED AND INDUCED FLOW' has been carried
out by Mr. Amit Kumar Singh under my supervision and
has not been submitted elsewhere for the award of a degree.

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Nov. 30, 1989

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# NOMENCLATURE

$C_1$ — Cost of drift loss, paise per hour

$C_2$ — Cost of power loss, paise per hour

CDE — Concrete drift eliminators

COP — Coefficient of performance

$g$ — Acceleration due to gravity, $ms^{-2}$

$h$ — Pressure drop in mm of water

$\dot{m}_a$ — Mass flow rate of air kg.min$^{-1}$

$m_d$ — Specific drift loss, kgw/kgda

$M_d$ — Rate of drift loss, kg h$^{-1}$

$m_e$ — Specific evaporation loss, kgw/kgda

$\dot{m}_w$ — Water mass flow rate, kg min$^{-1}$

$n$ — Number of stages

$p$ — Static pressure at a point, mm of water

$p_d$ — Discharge pressure of compressor, bar

$p_s$ — Suction pressure of compressor, bar

$P_{com}$      Power input to the compressor, W

$P_{FD}$      Power input to FD fan, kW

$P_{ID}$      Power input to ID fan, kW

$t$      Dry Bulb Temperature (DBT), $^{o}C$

$t^{*}$      Wet Bulb Temperature (WBT), $^{o}C$

$T_1$      Condenser inlet temperature, $^{o}C$

$T_2$      Condenser outlet temperature, $^{o}C$

$V_{vol}$      Volumetric discharge of air, $m^3 \ h^{-1}$

$V$      Supply voltage, V

$w$      Specific Humidity, kgw/kgda

$w_1$      Ambient air specific humidity, kgw/kgda

$w_2$      Specific Humidity of the leaving air without duct heater, kgw/kgda

$w_3$      Specific Humidity of the leaving air with duct heater, kgw/kgda

WDE      Wooden drift eliminators

$\rho_w$      Density of water, $kg/m^3$

$\Delta p$      Total pressure drop, mm of $H_2O$

$\theta$      Inclination Angle, degree

# ABSTRACT

A systematic experimental study was taken up on Drift
eliminators used in Evaporative Condensers to determine the
pressure drop across them, drift loss and its effect on system
performance. The wooden and concrete drift eliminator plates
were made and mounted in a frame which comprised a stage of
the drift eliminators. Several stages like this were made
for the current study. The experiments were conducted first
with a single stage wooden drift eliminators (WDE) and then
with double stages. The orientation of drift eliminator plates
was varied from $15^{\circ}$ to $90^{\circ}$ for one single fan speed. To alter
the fan speed, the supply voltage was varied from 160V to 230V
AC. Pressure drop and drift loss data were collected with FD
and ID fans for various orientations of drift eliminator plates
and for a range of fan speed. The above sets of experiments
were repeated for single stage and double stages of concrete drift
eliminators (CDE) and all the required data were collected.

Based upon the cost analysis, the variation of the
pressure drop and the drift loss suggested an optimum angle
of inclination of the plates. The optimum value depends upon

the number of stages used at a time. As the number of stages goes up, the optimum value of θ moves towards higher side.

The COP of the system increased with increase in fan speed and angle of inclination and decreased with increase in the number of stages. The use of FD fan resulted in better performance than that of ID fan.

# CHAPTER 1

## INTRODUCTION

### 1.1    GENERAL BACKGROUND

In evaporative cooling, water is sprayed into a flowing air media and the temperature of the water is reduced because of the evaporation of water and subsequent absorption of latent heat by the air stream.  Typical industrial applications where this principle is used are the cooling towers in power plant or central air conditioning systems, evaporative coolers for process industries and desert coolers.  In case of evaporative condensers, the water which is sprayed over the condenser tubes takes away some heat and secondly the air whose temperature is slightly increased due to the evaporation of water droplets, also removes some heat from the condenser tubes.  The heated water past the condenser is cooled by the process of adiabatic saturation.

In recent years, the demand for industrial water has increased and at the same time, the sources of raw water are limited.  To tackle this problem, we need an efficient mechanical system to retain the water since the spray or the water distribution system turns the process water into light droplets before they are cooled by the air stream.  As the air moves counter to

or cross-wise to the flow of water, it will pick up much of the mist and droplets and carry them with the air flow out of the tower or the evaporative condenser. This is known as drift loss. One way of tackling this problem of drift loss is by providing the drift eliminators above the condenser coil in an evaporative condenser and above the spray system in a cooling tower. The drift eliminators are typically rectangular plates which are set at the desired angle to the air flow direction over the whole outlet area of the air flow passage. By adjusting the angles of these plates in the flow direction suitably, most of the water droplets can be made to fall back into the sump, thereby reducing the drift loss.

Cooling tower performance, i.e., heat removal, is a balance between water flow and air volume, therefore drift eliminators are normally designed to be efficient through a calculated range of air flow. Too great an air speed can result in excessive drift loss of water from the tower,while poorly designed eliminators will adversely affect the performance of the unit. Thus drift eliminator effectiveness is an essential aspect of cooling tower design for many reasons, among than a few are:

1.      conservation of water;

2.      retention of chemicals used for the treatment in the
        water sump.

3.      prevention of staining by chemical additives eg.
        chromates etc.

4.      avoiding fan blades corrosion in case of induced draft.

5.      avoidance of violation of local area environmental
        protection regulations.

The present study concerns with the performance
characteristics of multistage wooden as well as concrete
drift eliminators with FD and ID fans separately; for an
evaporative condenser and its effect on the performance of
a refrigeration system.

## 1.2     LITERATURE SURVEY:

The principles of evaporative condenser has been
known for a long time. Because of the wide use that such a
condenser was put to in industrial applications, most books
on Refrigeration and Airconditioning mention about evaporative
condenser (Stoecker, 1982; Prasad, 1985). Another common
equipment which makes use of the principle of evaporative
cooling is the cooling tower.

Evaporative condenser may be preferred over cooling towers for small capacities but for larger capacities there is no other alternative but to use a cooling tower if conservation of water is very important. In the former as mentioned earlier the condensation takes place because of the heat extraction by the cold water stream falling over the tubes as well as by the relatively cold air stream. In the latter though, only the cooling of the water takes place inside the cooling tower by the process of adiabatic saturation, but the actual heat transfer from the condenser takes place away from the tower through indirect contact (Perry, 1973;Goodman, 1938).

Evaporative cooling with water recycling is a viable method of making efficient use of cooling water. Another method which finds use in small scale application is purely the cooling of condenser by air. The lowest condenser temperature, achievable in this fashion is equal to the Dry Bulb Temperature (DBT) of the entering air. But, by using the evaporative cooling process, the lowest condenser temperature can be brought close to the Wet Bulb Temperature (WBT) of the entering air. Because the wet bulb temperature is always less than or equal to dry bulb temperature, the refrigeration system can be operated with lower condensing temperature thereby leading to a higher COP, of the cycle.This can enhance the

cycle efficiency considerably particularly, in a non-humid hot climate, i.e., in deserts (Perry, 1973; Threlkeld, 1962).

In evaporative cooling, the heat-transfer process involves (1) latent heat transfer owing to vapourisation of a small portion of the water and (2) sensible heat transfer owing to the difference in temperature of water and air. Approximately 80% of this heat is due to latent heat and 20 percent to sensible heat. Theoretical possible heat removal per unit quantity of air circulated through an evaporative cooling equipment depends on the temperature and moisture content of air. An indication of the moisture content of the air is its wet-bulb temperature which is the lowest theoretical temperature to which the water can be cooled. Practically, the cold-water temperature approaches it, but does not equal the air wet-bulb temperature in the evaporative cooling process; this is because it is not possible to contact all the water with fresh air as the water drops through the evaporative condenser. So the important factors for an efficient evaporative cooling process are air to water contact (residence) time, and breakup of water into droplets (Boelter, 1939 ).

One of the problems associated with the evaporative cooling process is the carry over of water droplets with the air flow which is formed as 'drift loss'. This drift loss is

undesirable as it is a direct loss of water to the atmosphere. Particularly in places where there is a tremendous scarcity of water, this may result in a very big expense. The drift loss is also undesirable because of setting of water droplets on any electrical equipment may prove to be hazardous. Also in the case of Induced Draft Fan the water particles may intensify the corrosion of the blades thereby reducing the life of the fan. For these reasons, the thermal pollution created by the transported water droplets should be reduced.

Studies have been made on cellular drift eliminators for cooling towers. The material used for construction of cellular drift eliminator is neoprene-asbestos. A cellular drift eliminator installtion adds to the efficiency of the operation by stopping the air from making heavy turns arround herringbone, and also collects any droplets which are carried away by outgoing air. An additional bonus of neoprene-asbestos material is that it provides a fire proof barrier. Thus, a match, lighted cigrette, or burning rag from an incinerator that may land on top of the drift eliminators when tower is not in operation will not start a conflagration because the fire will extinguish itself on the asbestos (Burger, 1975).

Although evaporative condensers have been in use for a long time, till to date systematic studies have not been reported on the efficient use of drift eliminators in them. Details of the reduction in Drift Loss and the associated increase in power consumption while drift eliminators are used, have not been analysed. The present work attempts to investigate some of these aspects of the drift eliminator characteristics.

## 1.3    SCOPE OF THE PRESENT WORK:

The objectives of this research project are as follows

1.    Determination of the 'Pressure Drop' and 'Drift Loss' for one and two stages of the 'Drift Eliminators' for various orientations of the drift eliminator plates made of wood and concrete, in an evaporative condenser.

2.    Studying the variation of ' Pressure Drop' across the drift eliminators and 'Drift Loss' for various air flow rates through the evaporative condenser.

3.    Determination of an optimum angle of orientation for drift eliminators plates for the given capacity of the condenser and given geometry of the plates.

4.    Studying the effect of drift eliminators on the performance of a refrigeration system.

5.      Comparative study on the use of ID and FD fans
        in an evaporative cooling system.

        For the data in this study, the refrigeration system
capacity, water circulation rate and the type of spray system
were kept unchanged.


1.4     ORGANIZATION OF THE THESIS:

        Chapter two describes the experimental methodology
including the details of instrumentation and the medhod of
estimating the drift loss.  Chapter three describes the drift
loss, pressure drop and refrigeration system performance
data collected during the course of this investigation.
Discussion of the results including the main conclusions and
suggestions for future resrarch are also given.

# CHAPTER 2

## EXPERIMENTAL METHODOLOGY

### 2.1 TEST RIG:

The complete test rig consists of an evaporative condenser using a forced draft as well as an induced draft fan. The induced draft fan is mounted on a raised platform on the top of the test rig. The drift eliminators used are of two kinds; namely wooden and those made of concrete. The evaporative condenser has been installed as one of the components of a R-22 refrigeration system. The main components of the Test Rig are shown in Figure 2.1. Their description and the specifications are given below:

1. REFRIGERATION SYSTEM: Consists of the following

    a. Compressor:

    It's a SHRIRAM 1.5 ton capacity R-22 compressor.

    Specification                                : 1622

    Supply Voltage = 220V AC, Amps  =  12.2, RPM = 2850,

    Evaporating Temperature          =   $7.2^{\circ}C$,

    Condensing Temperature           =  $55^{\circ}C$

| | |
|---|---|
| Ambient Temperature | = 35°C, |
| Compressor Suction Gas Temperature | = 35°C, |
| Suction Pressure | = 5.25 bar (75.95 PSIG) |
| Discharge Pressure | =21.467 bar |
| | or |
| | (300.54 PSIG), |

b. <u>Condenser</u>:

Designed for a capacity of 1.5 ton. It is made of 3/8" (9mm) copper tube of length 23.10 m.

c. <u>Capillary Tube</u>:

It was provided for a rated capacity of 1.5 ton as an expansion device. The capillary tube length in 1.50 m.

d. <u>Evaporator (Cooling Coil)</u>:

The evaporator is 1.5 ton finned unit. A fan was installed at the back of this unit to blow ambient air for better heat transfer to the coil.

2. MAIN CHAMBER

Main chamber consists of (1) a rectangular box
(1.00m x 0.90 x 1.40 m)   (2) a drift eliminator
chamber (1.00 m x 0.52m x 0.52 m) having top portion
taperred and (3) a heater box (0.45m x 0.45 m x 0.45m).
The main chamber is made of angle iron frame and
aluminium sheet.  The Drift Eliminator chamber houses
drift eliminators supported on angles.  The heater
box houses 6 kW capacity finned electric duct  heater
on top of drift eliminator chamber.  The bottom portion
of the main chamber was used as a water sump as shown
in Figure 2.1.

3. DUCTS:

A main duct was connected at the top of the heater box
to the inlet of ID fan mounted on a raised plat-form
by the side of main chamber.  The discharge duct was
connected at the out let of ID fan to carry the
discharge air out of the room.  The dimensions of dis-
charge duct are 0.410m x 0.320m x 2.26m and was equipped
with a damper to control the flow in the sample duct.
Sample duct is of dimensions 0.18m x 0.18m x 1.20 m
which is used to measure the psycorometric condition
of air at its outlet.  The main duct has two parts

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1. Evaporator
2. Compressor
3. Condenser
4. Cap. Tube
5. Fan
6. Pump
7. Rotameter
8. Spray System
9. Sump
10. Drift Eliminators
11. FD Fan
12. Heater
13. Main Duct
14. ID Fan
15. Damper
16. Sample Duct

Fig. 2.1
Schematic Diagram of the Test Rig

(1) a 90° duct of cross section area 0.46m x 0.28m and (2) a connecting duct of rectangular cross-section of 0.46m x 0.28m at one end and a circular cross-section of dia 36 cm at the other end. The duct is 1.30 m long.

4.    DRIFT ELIMINATORS:

These were provided to reduce the drift loss. In the present case we have used two types of drift eliminators (1) Wooden drift eliminators(WDE) and (2) concrete drift eliminators (CDE). In case of wooden drift eliminators each strip can be rotated from 0° to 180°. For concrete drift eliminators a separate rotating device was used to change the orientation.

(a)    Wooden Drift Eliminators:

Specification : Area of box 0.495m x 0.95 m

| Thickness of each strip | = 13 mm |
|---|---|
| Number of strips in single stage | = 9 |
| Width of the each strip | = 46 mm |

(b)    Concrete Drift Eliminators:

| Thickness of each drift eliminator | = 25 mm |
|---|---|
| Length of each drift eliminator | = 900 mm |
| Width of each drift eliminator | = 50 mm |
| Number of drift eliminators in stage | = 7 |

5.   CIRCULATING WATER PUMP:

It was installed to provide the spray water over the
condenser coil through a sprayer consisting of 4 rows
of 1/2" conduite pipe with holes of size 2 mm dia.
The pump specifications are as follows

HP    = 2 , 3 $\phi$, Volt. 230/240

RPM   = 2840  ,  Amp  = 2.5

Make  = Alfa Laval

Type  = EM 1D 111B

6.   I.D. FAN AND F.D. FAN:

F.D. fan and motor assembly was installed on a concrete
foundation specially made for it by 6 foundation bolts.

Specifications:     Fan discharge 89.9m$^3$/min, static

pressure 13.5 mm $H_2O$   250 mm diameter,

Motor o.75 hp., 1420 rpm, 5.2 amp.

230 Volt., single phase. (Premier

Corporation India Ltd. Coimbotore).

I.D. fan specifications are same that of F.D. fan. I.D. fan
is mounted on a raised platfrom  supported on two 'L'
shape grounded 4" C.S.pipes.

## 2.2    INSTRUMENTATION USED:

Instruments were used to measure pressure and temperatures at different points in the refrigeration system. The pressure drop across the drift eliminators was measured by using a manometer directly calibrated in inches of water column. The specifications of various gauges used are given below.

a.    **Compressor Suction Side:**

| | |
|---|---|
| Pressure Gauge Range | 0 - 180 PSIG |
| Temperature | Copper-constanton themocouple. |

b.    **Compressor Discharge Side:**

| | |
|---|---|
| Pressure Gauge Range | 0 - 300 PSIG |
| Temperature | Copper-Constanton themocouple |

c.    **Condenser Outlet Side:**

| | |
|---|---|
| Temperature | Copper-Constanton themocouple |

d.    A vacuum pump was used to evacuate the refrigeration system before charging the refrigerant.

e. In order to determine the rate of water spray
over the condenser coil, a Rotameter (0-5 gpm,specific
gravity 1.0) was used to measure the circulating
water flow rate. For a check a water tank of
known volume capacity and a stop watch for
measuring the time were also used to measure
the flow rate of sprayed water on condenser.

f. The psychrometric data of the air entering and
leaving the evaporative condenser were measured
using an ordinary psychometer ( DBT -20 to 50$^{o}$C,
WBT -20$^{o}$C + 50$^{o}$C). To be able to measure these
data for the leaving air, the damper of the main
duct was closed partially so that the sufficient
amount of air could flow through the sample duct.
One Hygro-Thermograph ( M 594 Mfd The Bendix
Corporation, Maryland U.S.A.) was also used to
measure DBT and Relative Humidity directly.

g. Velocity of air was measured using a Vane
Anemometer ( 0 to 1 x 10$^{5}$m, wind speed 1 to 15 m/s,
OTA KEIKI SEISAKUSHO, JAPAN) at the inlet duct to
the F.D. fan.

h.   The power input to the compressor, FD and ID fans
     was measured by using Wattmeters.

| Wattmeter ( Compressor)- | Make | : Nippen |
|---|---|---|
| | Class | : (1.5) I.S. 1248-68 |
| | Model | : SF-144P-1EW |
| | Amp. | : 20 A |
| | Voltage | : 250 V |
| | Range | : 0- 1500 W |
| Wattmeter (FD and ID fan) | Make | : Toshniwal |
| | Range | : 0-2.5 kW |
| | Least count | : 0.05 kW |

i.   The supply voltage to the FD and ID fans was varied
     from 160 V to 230V bt using a Dimmerstat (0-270 Volts,
     max. load 15 amps, Mfd Automatic Electric Private
     Ltd- Bombay).

j.   The pressure drop across the drift eliminators
     was measured using a Manometer (Range - 0.1" to
     +1.0",Least count-0.02", Make-Dwyer, Sp. gravity
     of oil - 0.826).

k.   The temperatures sensed by thermocouples were read
     form an electronic temperature recorder (Honeywell
     Temperature Recorder Electronik-15, Range-0-300°F).

## 2.3 EXPERIMENTAL PROCEDURE:

Various steps of the experimental procedure are outlined below:

1. Before starting the experimental run, all the electrical connections, water connections were checked to ensure safety of the setup.

2. Wooden drift eliminators which were set $90^o$ to the horizontal were solid into the drift eliminator box.

3. The FD fan was run at 230V AC.

4. The pump was started for water supply to the spray system.

5. The refrigeration cycle was switched on and the pressure and temperatures at different points in the cycle were recorded. They were checked for not crossing the upper set limits.

6. After 15 minutes or so when the system was stablised, the suction and discharge pressures of the compressor, various temperatures of the refrigeration cycle, pressure drop across the drift eliminators using manometer, power consumed by the FD fan and compressor was recorded. The DBT and WBT were recorded for entering air at inlet to the main chamber and at the outlet of the sample duct.

7.     The heater (6 kW) mounted on top of the drift eliminator box was switched on for evaporation of water droplets carried away with the out going air.

8.     After about 15 minutes the readings were repeated for DBT and WBT at the out let of sample duct. Readings were also recorded by Hydrograph as a check. After the readings were taken heater was put off untill the next set of readings.

9.     Above set of readings were repeated for inclination angles of $60^{\circ}$, $45^{\circ}$, $30^{\circ}$ and $15^{\circ}$.

10.     Similarly, steps 2 to 8 were repeated for the two stages of wooden drift eliminators with FD fan.

11.     Then the ID fan was connected to the system and the experiment was repeated first.with single stage and then with double stages of wooden drift eliminators.

12.     All the above data were repeated for concrete drift eliminators with FD and ID fans (of course one at a time).

13.     Amount of water sprayed and also make up water were measured.

14.     Measure the barometric pressure and ambient temperature.

The damper of the discharge duct was kept partially closed throughout the experimental runs .

## 2.4    ESTIMATION OF DRIFT LOSS:

Psychrometric data were measured for the entering air and air leaving through the sample duct without and with heater on.  The damper of the discharge duct was kept partially closed.

### 1.    MEASUREMENT OF EVAPORATION LOSS:

This is the amount of water loss during the cooling process.  A simple mass balance of dry air and water over the evaporative condenser is given by

$$\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a \text{ (say)}$$

$$\dot{m}_{a_1} w_1 + \dot{m}_e + \dot{m}_d = \dot{m}_{a_2} w_2 + \dot{m}_d$$

Where, $\dot{m}_{a_1}$, $\dot{m}_{a_2}$ = mass of dry air entering and leaving the evaporative condenser

$t_1, t_2$ = dry bulb temperature (DBT) of the entering and leaving air.

$\dot{m}_e$ = rate of evaporation of water (i.e. evaporation loss)

$\dot{m}_d$ = drift loss

From equations (2.4.1) and (2.4.2) we get

$$\dot{m}_e = \dot{m}_a (w_2 - w_1)$$

$w_1$ and $w_2$ can be easily determined knowing the DBT and WBT of the entering and leaving air. The mass flow rate of the dry air can be calculated knowing the discharge rating of the fan (Das, 1988).

2.    MEASUREMENT OF THE DRIFT LOSS:

In order to measure $m_d$, the heater was switched on so that all the drift coming past the drift eliminators could be evaporated. This will change the quality of out going air. A simple mass balance over the evaporative condenser yields.

$$\dot{m}_a \, w_1 + \dot{m}_e + \dot{m}_d = \dot{m}_a \, w_3 \qquad (2.4)$$

or

$$\dot{m}_e + \dot{m}_d = \dot{m}_a \, (w_3 - w_1) \qquad (2.5)$$

Substituting from equation (2.4.3) into equation (2.5) yields

$$\dot{m}_d = \dot{m}_a \, (w_3 - w_2) \qquad (2.6)$$

and

$$\frac{\dot{m}_d}{\dot{m}_e} = \frac{w_3 - w_2}{w_2 - w_1} \qquad (2.7)$$

$\dot{m}_e$ and $\dot{m}_d$ can be easily determined by measuring the psychrometric data of the entering and the leaving air streams (Das, 1988).

# CHAPTER 3

## RESULTS AND DISCUSSION

In order to determine the drift loss from a varying number of drift eliminator stages and the pressure drop across them, experiments were conducted according to the precedure outlined in section 2.3.

## 3.1 DRIFT LOSS:

The angle of inclination ($\theta$) was varied from $15^\circ$ to $90^\circ$ and for any given set of data, the drift eliminators were set at a particular angle. The number of stages used were two or one at a time. The dry bulb and wet bulb temperatures were measured for the air entering and leaving the evaporative condenser without and with duct heater. The psychrometric data of the moist air recorded during the experiments are given in Tables 3.1, 3.2, 3.3 and 3.4. Table 3.1 shows data for FD fan using single stage and double stages concrete drift eliminators. Table 3.2 shows data for ID fan using single stage and double stage concrete drift eliminators. Table 3.3 shows data for FD fan using single stage and double stage wooden drift eliminators while Table 3.4 shows data for ID fan

using single and double stage wooden drift eliminators. The
evaporative $(m_e)$ and drift $(m_d)$ losses are computed and shown
in Tables 3.5, 3.6, 3.7 and 3.8. It can be seen from these
tables that the sp. drift loss increases if n decreases or
$\theta$ increases. This is basically due to the fact that in either
of the two cases (i.e. decreasing n or increasing $\theta$), the net
static pressure available for the flow increases which results
in a higher volumetric discharge. The greater discharge of
air brings larger amount of air indirect contact with water
resulting in a larger value of $m_e$.

The specific drift loss, $m_d$ is plotted versus $\theta$ for
various supply voltages i.e., 230V, 200V, 160V. This is shown
in Figures 3.1 through 3.6 covering single and double stages
of both concrete and wooden drift eliminators for FD fan as
well as ID fans. The trend of these curves is similar. As
$\theta$ increases, the drift loss $(m_d)$ increases, but it decreases
with increasing number of stages (i.e. double stage). The
drift loss as expected reaches a maximum for $\theta = 90^{\circ}$. As the
fan RPM increases (with increase in supply voltage), the drift
loss also goes up as shown in the figures mentioned above.
It is seen that drift loss is more with FD fan than with ID fan
of same capacity (see Figures 3.1 and 3.5). It is also observed
from the data that the drift loss while using concrete drift
eliminators is 20-25% less than that obtained using wooden drift

## Table 3.1

### Psychrometric Data of Moist Air Entering and Leaving the Evaporative Condenser

#### for FD fan with CDE

| Supply Voltage V (volts) | Inclination Angle θ (degree) | No. of stages n | Entering Air | | | Discharge Air without Heater | | | Discharge Air with Heater on | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | DBT $t_1$ (°C) | WBT $t_1^*$ (°C) | Humidity Ratio $w_1$ (kgw/kgda) | DBT $t_2$ (°C) | WBT $t_2^*$ (°C) | Humidity Ratio $w_2$ (kgw/kgda) | DBT $t_3$ (°C) | WBT $t_3^*$ (°C) | Humidity Ratio $w_3$ (kgw/kgda) |
| 230 | 15 | 1 | 29.8 | 26.9 | $21.0 \times 10^{-3}$ | 30.6 | 28.7 | $24.4 \times 10^{-3}$ | 34.1 | 29.7 | $24.8 \times 10^{-3}$ |
| | | 2 | 32.8 | 27.5 | $21.3 \times 10^{-3}$ | 32.7 | 29.0 | $25.5 \times 10^{-3}$ | 36.1 | 30.3 | $25.8 \times 10^{-3}$ |
| | 45 | 1 | 30.0 | 26.9 | $21.0 \times 10^{-3}$ | 31.1 | 29.1 | $24.8 \times 10^{-3}$ | 34.4 | 29.8 | $25.8 \times 10^{-3}$ |
| | | 2 | 30.4 | 26.3 | $19.8 \times 10^{-3}$ | 30.4 | 28.3 | $24.3 \times 10^{-3}$ | 34.4 | 29.8 | $25.4 \times 10^{-3}$ |
| | 60 | 1 | 30.0 | 27.1 | $21.6 \times 10^{-3}$ | 31.0 | 29.2 | $25.0 \times 10^{-3}$ | 34.4 | 29.8 | $25.8 \times 10^{-3}$ |
| | | 2 | 30.7 | 26.6 | $20.4 \times 10^{-3}$ | 30.9 | 28.6 | $24.6 \times 10^{-3}$ | 34.7 | 29.7 | $25.6 \times 10^{-3}$ |
| | 90 | 1 | 29.8 | 26.8 | $21.0 \times 10^{-3}$ | 30.5 | 28.7 | $24.4 \times 10^{-3}$ | 34.3 | 30.7 | $26.6 \times 10^{-3}$ |
| | | 2 | 31.0 | 26.9 | $20.6 \times 10^{-3}$ | 31.3 | 28.8 | $24.1 \times 10^{-3}$ | 34.8 | 30.7 | $26.6 \times 10^{-3}$ |
| | 15 | 1 | 32.1 | 26.9 | $20.2 \times 10^{-3}$ | 31.4 | 28.9 | $24.2 \times 10^{-3}$ | 34.8 | 29.6 | $24.5 \times 10^{-3}$ |
| | | 2 | 33.3 | 27.6 | $21.2 \times 10^{-3}$ | 32.2 | 29.5 | $25.0 \times 10^{-3}$ | 35.7 | 30.3 | $25.4 \times 10^{-3}$ |

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 200 | 45 | 1 | 31.8 | 26.5 | $19.6 \times 10^{-3}$ | 31.2 | 28.4 | $23.6 \times 10^{-3}$ | 34.9 | 29.3 | $23.9 \times 10^{-3}$ |
| | | 2 | 33.5 | 27.7 | $21.2 \times 10^{-3}$ | 32.2 | 29.4 | $24.4 \times 10^{-3}$ | 35.8 | 30.3 | $25.0 \times 10^{-3}$ |
| | 60 | 1 | 31.4 | 26.7 | $20.2 \times 10^{-3}$ | 30.7 | 28.5 | $23.6 \times 10^{-3}$ | 34.6 | 29.3 | $24.8 \times 10^{-3}$ |
| | | 2 | 33.4 | 27.8 | $21.4 \times 10^{-3}$ | 32.3 | 28.6 | $24.4 \times 10^{-3}$ | 35.9 | 30.3 | $25.3 \times 10^{-3}$ |
| | 90 | 1 | 30.6 | 26.6 | $20.4 \times 10^{-3}$ | 30.2 | 28.5 | $24.0 \times 10^{-3}$ | 31.8 | 28.7 | $26.0 \times 10^{-3}$ |
| | | 2 | 32.8 | 27.3 | $20.8 \times 10^{-3}$ | 33.0 | 28.6 | $23.0 \times 10^{-3}$ | 34.8 | 29.8 | $24.7 \times 10^{-3}$ |
| 160 | 15 | 1 | 32.1 | 26.9 | $20.1 \times 10^{-3}$ | 31.3 | 28.8 | $24.3 \times 10^{-3}$ | 34.6 | 29.5 | $24.5 \times 10^{-3}$ |
| | | 2 | 33.5 | 27.6 | $21.2 \times 10^{-3}$ | 32.1 | 29.4 | $25.0 \times 10^{-3}$ | 35.7 | 30.3 | $25.2 \times 10^{-3}$ |
| | 45 | 1 | 31.3 | 26.5 | $19.8 \times 10^{-3}$ | 31.0 | 28.4 | $23.0 \times 10^{-3}$ | 34.7 | 29.2 | $23.4 \times 10^{-3}$ |
| | | 2 | 33.5 | 27.7 | $21.0 \times 10^{-3}$ | 32.2 | 29.6 | $25.0 \times 10^{-3}$ | 35.8 | 30.4 | $25.5 \times 10^{-3}$ |
| | 60 | 1 | 31.4 | 26.7 | $20.3 \times 10^{-3}$ | 30.6 | 28.5 | $23.9 \times 10^{-3}$ | 34.6 | 29.2 | $24.4 \times 10^{-3}$ |
| | | 2 | 33.4 | 28.2 | $22.0 \times 10^{-3}$ | 32.3 | 29.5 | $24.0 \times 10^{-3}$ | 35.6 | 29.8 | $24.7 \times 10^{-3}$ |
| | 90 | 1 | 30.5 | 26.5 | $20.4 \times 10^{-3}$ | 30.0 | 28.4 | $23.8 \times 10^{-3}$ | 32.2 | 29.7 | $24.8 \times 10^{-3}$ |
| | | 2 | 32.8 | 27.6 | $214 \times 10^{-3}$ | 32.0 | 29.6 | $24.6 \times 10^{-3}$ | 34.8 | 29.6 | $24.6 \times 10^{-3}$ |

## Table 3.2

Psychrometric Data of Moist Air Entering and Leaving the Evaporative Condenser For FD fan with CDE

| Supply Voltage v (volts) | Inclination Angle θ (degree) | No.of stages n | Entering Air | | | Discharge Air Without Heater | | | Discharge Air with Heater on | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | DBT $t_1$ (°C) | WBT $t_1^*$ (°C) | Humidity Ratio $w_1$ (kgw/kgda) | DBT $t_2$ (°C) | WBT $t_2^*$ (°C) | Humidity Ratio $w_2$ (kgw/kgda) | DBT $t_3$ (°C) | WBT $t_3^*$ (°C) | Humidity Ratio $w_3$ (kgw/kgda) |
| | 15 | 1 | 29.8 | 26.9 | $21.2 \times 10^{-3}$ | 30.6 | 28.7 | $24.4 \times 10^{-3}$ | 34.1 | 29.7 | $24.6 \times 10^{-3}$ |
| | | 2 | 29.8 | 26.9 | $21.2 \times 10^{-3}$ | 32.5 | 29.0 | $25.0 \times 10^{-3}$ | 35.9 | 30.0 | $25.4 \times 10^{-3}$ |
| | 45 | 1 | 30.0 | 26.9 | $21.0 \times 10^{-3}$ | 30.0 | 28.5 | $24.0 \times 10^{-3}$ | 34.2 | 20.4 | $25.0 \times 10^{-3}$ |
| | | 2 | 30.0 | 26.9 | $21.0 \times 10^{-3}$ | 30.8 | 28.3 | $23.4 \times 10^{-3}$ | 34.4 | 29.8 | $24.0 \times 10^{-3}$ |
| | 60 | 1 | 28.8 | 26.9 | $21.6 \times 10^{-3}$ | 30.1 | 28.5 | $24.0 \times 10^{-3}$ | 33.7 | 30.0 | $25.6 \times 10^{-3}$ |
| 230 | | 2 | 28.8 | 26.9 | $21.6 \times 10^{-3}$ | 30.6 | 28.4 | $24.6 \times 10^{-3}$ | 33.5 | 30.5 | $25.0 \times 10^{-3}$ |
| | 90 | 1 | 29.8 | 26.8 | $21.0 \times 10^{-3}$ | 30.6 | 28.8 | $24.8 \times 10^{-3}$ | 32.4 | 21.0 | $26.0 \times 10^{-3}$ |
| | | 2 | 29.8 | 26.8 | $21.0 \times 10^{-3}$ | 31.0 | 28.8 | $24.1 \times 10^{-3}$ | 32.8 | 21.0 | $26.0 \times 10^{-3}$ |
| | 15 | 1 | 29.8 | 26.9 | $21.2 \times 10^{-3}$ | 30.6 | 28.7 | $24.6 \times 10^{-3}$ | 34.1 | 29.7 | $25.1 \times 10^{-3}$ |
| | | 2 | 33.0 | 27.6 | $21.2 \times 10^{-3}$ | 32.5 | 29.5 | $24.0 \times 10^{-3}$ | 35.8 | 30.0 | $24.9 \times 10^{-3}$ |

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 200 | 45 | 1 | 30.0 | 26.9 | $21.0 \times 10^{-3}$ | 30.7 | 28.9 | $23.4 \times 10^{-3}$ | 34.2 | 29.4 | $24.2 \times 10^{-3}$ |
| | | 2 | 30.4 | 26.3 | $20.0 \times 10^{-3}$ | 30.6 | 28.3 | $23.4 \times 10^{-3}$ | 34.4 | 29.1 | $24.9 \times 10^{-3}$ |
| | 60 | 1 | 28.8 | 26.9 | $21.6 \times 10^{-3}$ | 29.9 | 28.0 | $23.7 \times 10^{-3}$ | 34.6 | 29.9 | $25.2 \times 10^{-3}$ |
| | | 2 | 30.7 | 26.6 | $20.4 \times 10^{-3}$ | 30.8 | 28.5 | $23.8 \times 10^{-3}$ | 34.6 | 29.7 | $24.9 \times 10^{-3}$ |
| | 90 | 1 | 29.8 | 26.8 | $21.0 \times 10^{-3}$ | 30.5 | 28.6 | $24.2 \times 10^{-3}$ | 34.3 | 30.7 | $26.0 \times 10^{-3}$ |
| | | 2 | 31.0 | 26.9 | $20.6 \times 10^{-3}$ | 31.2 | 28.8 | $24.2 \times 10^{-3}$ | 34.8 | 30.8 | $25.8 \times 10^{-3}$ |
| | 15 | 1 | 32.1 | 26.9 | $20.2 \times 10^{-3}$ | 31.8 | 29.0 | $24.4 \times 10^{-3}$ | 35.0 | 29.8 | $24.8 \times 10^{-3}$ |
| | | 2 | 33.3 | 27.6 | $21.2 \times 10^{-3}$ | 32.3 | 29.0 | $24.7 \times 10^{-3}$ | 35.9 | 30.4 | $25.0 \times 10^{-3}$ |
| 160 | 45 | 1 | 31.8 | 26.5 | $19.6 \times 10^{-3}$ | 31.8 | 28.5 | $23.3 \times 10^{-3}$ | 35.0 | 29.5 | $24.0 \times 10^{-3}$ |
| | | 2 | 33.5 | 27.7 | $21.0 \times 10^{-3}$ | 32.2 | 29.4 | $25.1 \times 10^{-3}$ | 35.9 | 30.3 | $25.5 \times 10^{-3}$ |
| | 60 | 1 | 31.4 | 26.7 | $20.3 \times 10^{-3}$ | 31.0 | 28.7 | $24.2 \times 10^{-3}$ | 34.5 | 30.5 | $25.2 \times 10^{-3}$ |
| | | 2 | 33.4 | 27.8 | $21.4 \times 10^{-3}$ | 32.6 | 29.7 | $25.0 \times 10^{-3}$ | 26.0 | 30.5 | $25.8 \times 10^{-3}$ |
| | 90 | 1 | 30.6 | 26.6 | $20.4 \times 10^{-3}$ | 30.4 | 28.6 | $24.3 \times 10^{-3}$ | 34.9 | 29.5 | $26.0 \times 10^{-3}$ |
| | | 2 | 32.8 | 27.3 | $20.5 \times 10^{-3}$ | 32.6 | 28.5 | $25.6 \times 10^{-3}$ | 34.8 | 29.7 | $24.8 \times 10^{-3}$ |

## Table 3.3

**Psychrometric Data of the Moist Air Entering and Leaving the Evaporative Condenser**

**for FD fan with WDE**

| Supply Voltage v (Volts) | Inclination Angle θ (degree) | No.of stages n | Entering Air | | | Discharge Air Without Heater | | | Discharge Air with Heater on | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | DBT $t_1$ (°C) | WBT $t_1^*$ (°C) | Humidity Ratio $w_1$ (kgw/kgda) | DBT $t_2$ (°C) | WBT $t_2^*$ (°C) | Humidity Ratio $w_2$ (kgw/kgda) | DBT $t_3$ (°C) | WBT $t_3^*$ (°C) | Humidity Ratio $w_3$ (kgw/kgda) |
| | 15 | 1 | 30.0 | 27.0 | $20.8 \times 10^{-3}$ | 30.5 | 28.5 | $24.0 \times 10^{-3}$ | 32.0 | 29.0 | $25.0 \times 10^{-3}$ |
| | | 2 | 30.0 | 27.0 | $20.8 \times 10^{-3}$ | 30.5 | 28.0 | $24.0 \times 10^{-3}$ | 31.0 | 28.5 | $24.4 \times 10^{-3}$ |
| | 30 | 1 | 29.5 | 27.0 | $21.4 \times 10^{-3}$ | 29.6 | 28.3 | $23.8 \times 10^{-3}$ | 31.0 | 29.5 | $25.1 \times 10^{-3}$ |
| | | 2 | 29.5 | 27.2 | $21.6 \times 10^{-3}$ | 29.8 | 28.3 | $23.4 \times 10^{-3}$ | 32.0 | 29.8 | $26.0 \times 10^{-3}$ |
| | 45 | 1 | 30.5 | 27.2 | $21.4 \times 10^{-3}$ | 30.0 | 28.5 | $24.6 \times 10^{-3}$ | 31.0 | 30.0 | $26.6 \times 10^{-3}$ |
| | | 2 | 29.8 | 27.5 | $22.2 \times 10^{-3}$ | 29.6 | 28.6 | $24.6 \times 10^{-3}$ | 31.5 | 30.0 | $26.3 \times 10^{-3}$ |
| | 60 | 1 | 30.3 | 27.5 | $22.0 \times 10^{-3}$ | 30.5 | 29.0 | $24.8 \times 10^{-3}$ | 31.5 | 30.5 | $27.0 \times 10^{-3}$ |
| | | 2 | 30.0 | 27.2 | $21.0 \times 10^{-3}$ | 30.5 | 29.0 | $24.8 \times 10^{-3}$ | 31.5 | 30.3 | $26.8 \times 10^{-3}$ |
| | 90 | 1 | 30.5 | 27.0 | $21.6 \times 10^{-3}$ | 30.8 | 29.0 | $24.8 \times 10^{-3}$ | 31.8 | 30.6 | $27.4 \times 10^{-3}$ |
| | | 2 | 30.6 | 27.3 | $22.2 \times 10^{-3}$ | 30.5 | 29.0 | $24.7 \times 10^{-3}$ | 31.8 | 30.5 | $27.3 \times 10^{-3}$ |

Table 3.4

Psychrometric Data of Moist Air Entering and Leaving the Evaporative Condenser

for ID fan with WDE

| Supply Voltage V (volts) | Inclination Angle θ (degree) | No.of stages n | Entering Air | | | Discharge Air without Heater | | | Discharge Air with Heater on | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | DBT $t_1$ (°C) | WBT $t_1^*$ (°C) | Humidity Ratio $w_1$ (kgw/kgda) | DBT $t_2$ (°C) | WBT $t_2^*$ (°C) | Humidity Ratio $w_2$ (kgw/kgda) | DBT $t_3$ (°C) | WBT $t_3^*$ (°C) | Humidity Ratio $w_3$ (kgw/kgda) |
| 230 | 15 | 1 | 32.5 | 26.9 | $20.0 \times 10^{-3}$ | 32.3 | 29.0 | $24.2 \times 10^{-3}$ | 35.3 | 30.6 | $24.9 \times 10^{-3}$ |
| | | 2 | 32.2 | 26.6 | $19.7 \times 10^{-3}$ | 32.2 | 28.9 | $23.8 \times 10^{-3}$ | 35.4 | 30.1 | $25.0 \times 10^{-3}$ |
| | 30 | 1 | 30.7 | 27.3 | $21.3 \times 10^{-3}$ | 31.1 | 29.1 | $24.7 \times 10^{-3}$ | 34.7 | 30.0 | $25.9 \times 10^{-3}$ |
| | | 2 | 32.5 | 27.2 | $20.6 \times 10^{-3}$ | 31.0 | 28.8 | $23.6 \times 10^{-3}$ | 35.2 | 30.0 | $25.0 \times 10^{-3}$ |
| | 45 | 1 | 30.5 | 27.0 | $21.0 \times 10^{-3}$ | 31.0 | 28.9 | $24.4 \times 10^{-3}$ | 34.5 | 30.0 | $26.4 \times 10^{-3}$ |
| | | 2 | 32.0 | 26.7 | $20.0 \times 10^{-3}$ | 31.9 | 28.8 | $24.0 \times 10^{-3}$ | 35.0 | 29.8 | $25.9 \times 10^{-3}$ |
| | 60 | 1 | 29.5 | 27.3 | $22.0 \times 10^{-3}$ | 30.7 | 29.1 | $24.8 \times 10^{-3}$ | 34.2 | 30.0 | $26.8 \times 10^{-3}$ |
| | | 2 | 31.1 | 26.8 | $20.4 \times 10^{-3}$ | 31.3 | 28.7 | $24.0 \times 10^{-3}$ | 35.2 | 30.2 | $26.2 \times 10^{-3}$ |
| | 90 | 1 | 32.8 | 26.9 | $19.8 \times 10^{-3}$ | 32.4 | 28.9 | $23.8 \times 10^{-3}$ | 35.2 | 30.0 | $25.4 \times 10^{-3}$ |
| | | 2 | 32.8 | 26.9 | $19.8 \times 10^{-3}$ | 32.2 | 28.8 | $23.6 \times 10^{-3}$ | 35.0 | 30.8 | $26.0 \times 10^{-3}$ |

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 15 | 1 | 32.5 | 26.9 | $20.2 \times 10^{-3}$ | 32.3 | 29.0 | $24.0 \times 10^{-3}$ | 36.6 | 30.2 | $24.8 \times 10^{-3}$ |
| | | 2 | 32.2 | 29.2 | $23.4 \times 10^{-3}$ | 32.2 | 29.1 | $23.9 \times 10^{-3}$ | 36.3 | 30.2 | $25.0 \times 10^{-3}$ |
| | 30 | 1 | 30.8 | 27.3 | $21.4 \times 10^{-3}$ | 31.0 | 29.0 | $24.6 \times 10^{-3}$ | 34.8 | 30.3 | $26.2 \times 10^{-3}$ |
| | | 2 | 32.5 | 27.2 | $20.4 \times 10^{-3}$ | 31.8 | 29.0 | $24.3 \times 10^{-3}$ | 35.2 | 30.0 | $25.2 \times 10^{-3}$ |
| 200 | 45 | 1 | 30.5 | 27.4 | $22.0 \times 10^{-3}$ | 31.3 | 29.0 | $24.4 \times 10^{-3}$ | 34.8 | 30.4 | $25.6 \times 10^{-3}$ |
| | | 2 | 32.0 | 26.7 | $20.2 \times 10^{-3}$ | 31.8 | 28.9 | $24.0 \times 10^{-3}$ | 35.8 | 29.9 | $25.6 \times 10^{-3}$ |
| | 60 | 1 | 29.5 | 27.3 | $22.4 \times 10^{-3}$ | 31.1 | 29.1 | $24.8 \times 10^{-3}$ | 34.5 | 29.9 | $26.2 \times 10^{-3}$ |
| | | 2 | 31.0 | 26.7 | $20.6 \times 10^{-3}$ | 31.3 | 28.9 | $24.3 \times 10^{-3}$ | 35.0 | 29.8 | $26.2 \times 10^{-3}$ |
| | 90 | 1 | 32.5 | 26.9 | $20.4 \times 10^{-3}$ | 32.8 | 28.9 | $23.6 \times 10^{-3}$ | 34.4 | 29.9 | $25.6 \times 10^{-3}$ |
| | | 2 | 32.5 | 26.8 | $20.3 \times 10^{-3}$ | 32.9 | 28.8 | $23.4 \times 10^{-3}$ | 34.5 | 29.8 | $25.4 \times 10^{-3}$ |
| | 15 | 1 | 32.5 | 26.9 | $20.0 \times 10^{-3}$ | 32.3 | 29.0 | $24.2 \times 10^{-3}$ | 36.3 | 30.2 | $25.0 \times 10^{-3}$ |
| | | 2 | 32.2 | 26.6 | $19.8 \times 10^{-3}$ | 32.1 | 28.0 | $23.2 \times 10^{-3}$ | 36.4 | 30.1 | $23.8 \times 10^{-3}$ |
| | 30 | 1 | 30.7 | 27.3 | $21.5 \times 10^{-3}$ | 31.0 | 29.1 | $24.6 \times 10^{-3}$ | 34.7 | 30.2 | $25.4 \times 10^{-3}$ |
| | | 2 | 32.5 | 27.2 | $20.6 \times 10^{-3}$ | 31.5 | 28.8 | $24.2 \times 10^{-3}$ | 35.2 | 30.0 | $25.0 \times 10^{-3}$ |
| 160 | 45 | 1 | 30.5 | 27.4 | $21.8 \times 10^{-3}$ | 31.1 | 28.9 | $24.4 \times 10^{-3}$ | 34.5 | 30.1 | $25.4 \times 10^{-3}$ |
| | | 2 | 32.0 | 26.7 | $19.8 \times 10^{-3}$ | 31.8 | 28.8 | $24.0 \times 10^{-3}$ | 35.9 | 29.8 | $24.0 \times 10^{-3}$ |
| | 60 | 1 | 29.5 | 27.3 | $22.1 \times 10^{-3}$ | 30.8 | 28.1 | $23.8 \times 10^{-3}$ | 34.4 | 29.8 | $24.9 \times 10^{-3}$ |
| | | 2 | 31.1 | 26.8 | $20.6 \times 10^{-3}$ | 31.2 | 28.8 | $24.3 \times 10^{-3}$ | 34.1 | 29.8 | $25.7 \times 10^{-3}$ |
| | 90 | 1 | 32.8 | 26.9 | $20.0 \times 10^{-3}$ | 32.5 | 28.9 | $23.6 \times 10^{-4}$ | 34.2 | 29.9 | $25.3 \times 10^{-3}$ |
| | | 2 | 32.8 | 26.9 | $20.0 \times 10^{-3}$ | 32.5 | 28.8 | $23.4 \times 10^{-3}$ | 34.3 | 29.8 | $25.0 \times 10^{-3}$ |

## Table 3.5

### Drift Loss Data for FD fan with CDE

| Supply Voltage V Volts | Inclination Angle $\theta$ (degree) | No.of stages n | Evaporation loss $m_e$ (kgw/kgda) | Drift Loss $m_d$ (kgw/kgda) |
|---|---|---|---|---|
| 230 | 15 | 1 | $3.3 \times 10^{-3}$ | $0.4 \times 10^{-3}$ |
| | | 2 | $4.3 \times 10^{-3}$ | $0.3 \times 10^{-3}$ |
| | 45 | 1 | $3.8 \times 10^{-3}$ | $1.0 \times 10^{-3}$ |
| | | 2 | $4.5 \times 10^{-3}$ | $0.7 \times 10^{-3}$ |
| | 60 | 1 | $4.6 \times 10^{-3}$ | $1.1 \times 10^{-3}$ |
| | | 2 | $4.2 \times 10^{-3}$ | $1.0 \times 10^{-3}$ |
| | 90 | 1 | $3.4 \times 10^{-3}$ | $2.2 \times 10^{-3}$ |
| | | 2 | $3.5 \times 10^{-3}$ | $1.9 \times 10^{-3}$ |
| 200 | 15 | 1 | $4.0 \times 10^{-3}$ | $0.3 \times 10^{-3}$ |
| | | 2 | $3.8 \times 10^{-3}$ | $0.4 \times 10^{-3}$ |
| | 45 | 1 | $4.0 \times 10^{-3}$ | $0.6 \times 10^{-3}$ |
| | | 2 | $3.1 \times 10^{-3}$ | $0.6 \times 10^{-3}$ |
| | 60 | 1 | $3.6 \times 10^{-3}$ | $1.2 \times 10^{-3}$ |
| | | 2 | $3.6 \times 10^{-3}$ | $0.8 \times 10^{-3}$ |
| | 90 | 1 | $3.6 \times 10^{-3}$ | $2.0 \times 10^{-3}$ |
| | | 2 | $3.7 \times 10^{-3}$ | $1.7 \times 10^{-3}$ |

| 160 | 15 | 1 | $4.1 \times 10^{-3}$ | $0.2 \times 10^{-3}$ |
| | | 2 | $3.8 \times 10^{-3}$ | $0.2 \times 10^{-3}$ |
| | 45 | 1 | $3.2 \times 10^{-3}$ | $0.4 \times 10^{-3}$ |
| | | 2 | $4.4 \times 10^{-3}$ | $0.5 \times 10^{-3}$ |
| | 60 | 1 | $3.6 \times 10^{-3}$ | $0.5 \times 10^{-3}$ |
| | | 2 | $2.2 \times 10^{-3}$ | $0.7 \times 10^{-3}$ |
| | 90 | 1 | $3.4 \times 10^{-3}$ | $1.0 \times 10^{-3}$ |
| | | 2 | $4.2 \times 10^{-3}$ | $1.0 \times 10^{-3}$ |

Table 3.6

Drift Loss Data for ID fan with CDE

| Supply Voltage V Volts | Inclination Angle $\theta$ (degree) | No. of stages n | Evaporation loss $m_e$ (kgw/kgda) | Drift Loss $m_d$ (kgw/kgda) |
|---|---|---|---|---|
| 230 | 15 | 1 | $3.2 \times 10^{-3}$ | $0.6 \times 10^{-3}$ |
| | | 2 | $3.8 \times 10^{-3}$ | $0.4 \times 10^{-3}$ |
| | 45 | 1 | $3.0 \times 10^{-3}$ | $1.0 \times 10^{-3}$ |
| | | 2 | $2.4 \times 10^{-3}$ | $0.6 \times 10^{-3}$ |
| | 60 | 1 | $2.4 \times 10^{-3}$ | $1.6 \times 10^{-3}$ |
| | | 2 | $2.0 \times 10^{-3}$ | $1.4 \times 10^{-3}$ |
| | 90 | 1 | $2.8 \times 10^{-3}$ | $2.0 \times 10^{-3}$ |
| | | 2 | $3.6 \times 10^{-3}$ | $1.8 \times 10^{-3}$ |
| 200 | 15 | 1 | $3.2 \times 10^{-3}$ | $0.5 \times 10^{-3}$ |
| | | 2 | $2.8 \times 10^{-3}$ | $0.3 \times 10^{-3}$ |
| | 45 | 1 | $2.4 \times 10^{-3}$ | $0.8 \times 10^{-3}$ |
| | | 2 | $3.4 \times 10^{-3}$ | $0.5 \times 10^{-3}$ |
| | 60 | 1 | $1.6 \times 10^{-3}$ | $1.3 \times 10^{-3}$ |
| | | 2 | $3.4 \times 10^{-3}$ | $1.0 \times 10^{-3}$ |
| | 90 | 1 | $3.2 \times 10^{-3}$ | $1.8 \times 10^{-3}$ |
| | | 2 | $3.6 \times 10^{-3}$ | $1.6 \times 10^{-3}$ |

| 160 | 15 | 1 | $4.2 \times 10^{-3}$ | $0.4 \times 10^{-3}$ |
| | | 2 | $3.0 \times 10^{-3}$ | $0.3 \times 10^{-3}$ |
| | 45 | 1 | $3.7 \times 10^{-3}$ | $0.7 \times 10^{-3}$ |
| | | 2 | $4.0 \times 10^{-3}$ | $0.4 \times 10^{-3}$ |
| | 60 | 1 | $3.9 \times 10^{-3}$ | $0.4 \times 10^{-3}$ |
| | | 2 | $4.0 \times 10^{-3}$ | $1.0 \times 10^{-3}$ |
| | 90 | 1 | $3.9 \times 10^{-3}$ | $1.7 \times 10^{-3}$ |
| | | 2 | $2.8 \times 10^{-3}$ | $1.2 \times 10^{-3}$ |

Table 3.7

Drift Loss Data for ID fan with WDE

| Supply Voltage V Volts | Inclination Angle $\theta$ (degree) | No. of stages n | Evaporation loss $m_e$ (kgw/kgda) | Drift Loss $m_d$ (kgw/kgda) |
|---|---|---|---|---|
| 230 | 15 | 1 | $4.2 \times 10^{-3}$ | $1.7 \times 10^{-3}$ |
| | | 2 | $4.1 \times 10^{-3}$ | $1.2 \times 10^{-3}$ |
| | 30 | 1 | $3.4 \times 10^{-3}$ | $1.2 \times 10^{-3}$ |
| | | 2 | $3.6 \times 10^{-3}$ | $1.4 \times 10^{-3}$ |
| | 45 | 1 | $4.0 \times 10^{-3}$ | $1.9 \times 10^{-3}$ |
| | | 2 | $3.4 \times 10^{-3}$ | $2.0 \times 10^{-3}$ |
| | 60 | 1 | $2.8 \times 10^{-3}$ | $2.0 \times 10^{-3}$ |
| | | 2 | $3.6 \times 10^{-3}$ | $2.2 \times 10^{-3}$ |
| | 90 | 1 | $4.0 \times 10^{-3}$ | $2.6 \times 10^{-3}$ |
| | | 2 | $3.8 \times 10^{-3}$ | $2.4 \times 10^{-3}$ |
| | 15 | 1 | $3.8 \times 10^{-3}$ | $0.8 \times 10^{-3}$ |
| | | 2 | $2.4 \times 10^{-3}$ | $1.1 \times 10^{-3}$ |
| | 30 | 1 | $3.2 \times 10^{-3}$ | $1.6 \times 10^{-3}$ |
| | | 2 | $3.9 \times 10^{-3}$ | $0.9 \times 10^{-3}$ |

| | | | | |
|---|---|---|---|---|
| 200 | 45 | 1 | $2.4 \times 10^{-3}$ | $1.8 \times 10^{-3}$ |
| | | 2 | $3.8 \times 10^{-3}$ | $1.6 \times 10^{-3}$ |
| | 60 | 1 | $2.4 \times 10^{-3}$ | $1.4 \times 10^{-3}$ |
| | | 2 | $3.6 \times 10^{-3}$ | $1.4 \times 10^{-3}$ |
| | 90 | 1 | $3.2 \times 10^{-3}$ | $2.0 \times 10^{-3}$ |
| | | 2 | $3.1 \times 10^{-3}$ | $1.8 \times 10^{-3}$ |
| 160 | 15 | 1 | $4.2 \times 10^{-3}$ | $0.9 \times 10^{-3}$ |
| | | 2 | $2.4 \times 10^{-3}$ | $0.6 \times 10^{-3}$ |
| | 30 | 1 | $3.1 \times 10^{-3}$ | $0.8 \times 10^{-3}$ |
| | | 2 | $3.6 \times 10^{-3}$ | $0.8 \times 10^{-3}$ |
| | 45 | 1 | $2.6 \times 10^{-3}$ | $1.0 \times 10^{-3}$ |
| | | 2 | $4.2 \times 10^{-3}$ | $1.0 \times 10^{-3}$ |
| | 60 | 1 | $2.7 \times 10^{-3}$ | $1.1 \times 10^{-3}$ |
| | | 2 | $3.7 \times 10^{-3}$ | $1.4 \times 10^{-3}$ |
| | 90 | 1 | $3.6 \times 10^{-3}$ | $1.7 \times 10^{-3}$ |
| | | 2 | $3.6 \times 10^{-3}$ | $1.6 \times 10^{-3}$ |

## Table 3.8

### Drift Loss Data for FD fan with WDE

| Supply Voltage V Volts | Inclination Angle θ (degree) | No. of stages n | Evaporation loss $m_e$ (kgw/kgda) | Drift loss $m_d$ (kgw/kgda) |
|---|---|---|---|---|
| 230 | 15 | 1 | $3.2\times10^{-3}$ | $1.0\times10^{-3}$ |
| | | 2 | $3.2\times10^{-3}$ | $0.4\times10^{-3}$ |
| | 30 | 1 | $2.4\times10^{-3}$ | $1.4\times10^{-3}$ |
| | | 2 | $2.2\times10^{-3}$ | $1.2\times10^{-3}$ |
| | 45 | 1 | $3.2\times10^{-3}$ | $2.0\times10^{-3}$ |
| | | 2 | $2.4\times10^{-3}$ | $1.7\times10^{-3}$ |
| | 60 | 1 | $3.8\times10^{-3}$ | $2.2\times10^{-3}$ |
| | | 2 | $3.2\times10^{-3}$ | $1.9\times10^{-3}$ |
| | 90 | 1 | $3.2\times10^{-3}$ | $2.6\times10^{-3}$ |
| | | 2 | $2.5\times10^{-3}$ | $2.4\times10^{-3}$ |

Water flow rate = 60.66 kg/min.

Make-up water = 2.14 kg/min.

$(\times 10^{-3})$

3.0

2.0

1.0

$m_d$

230V
200V
160V

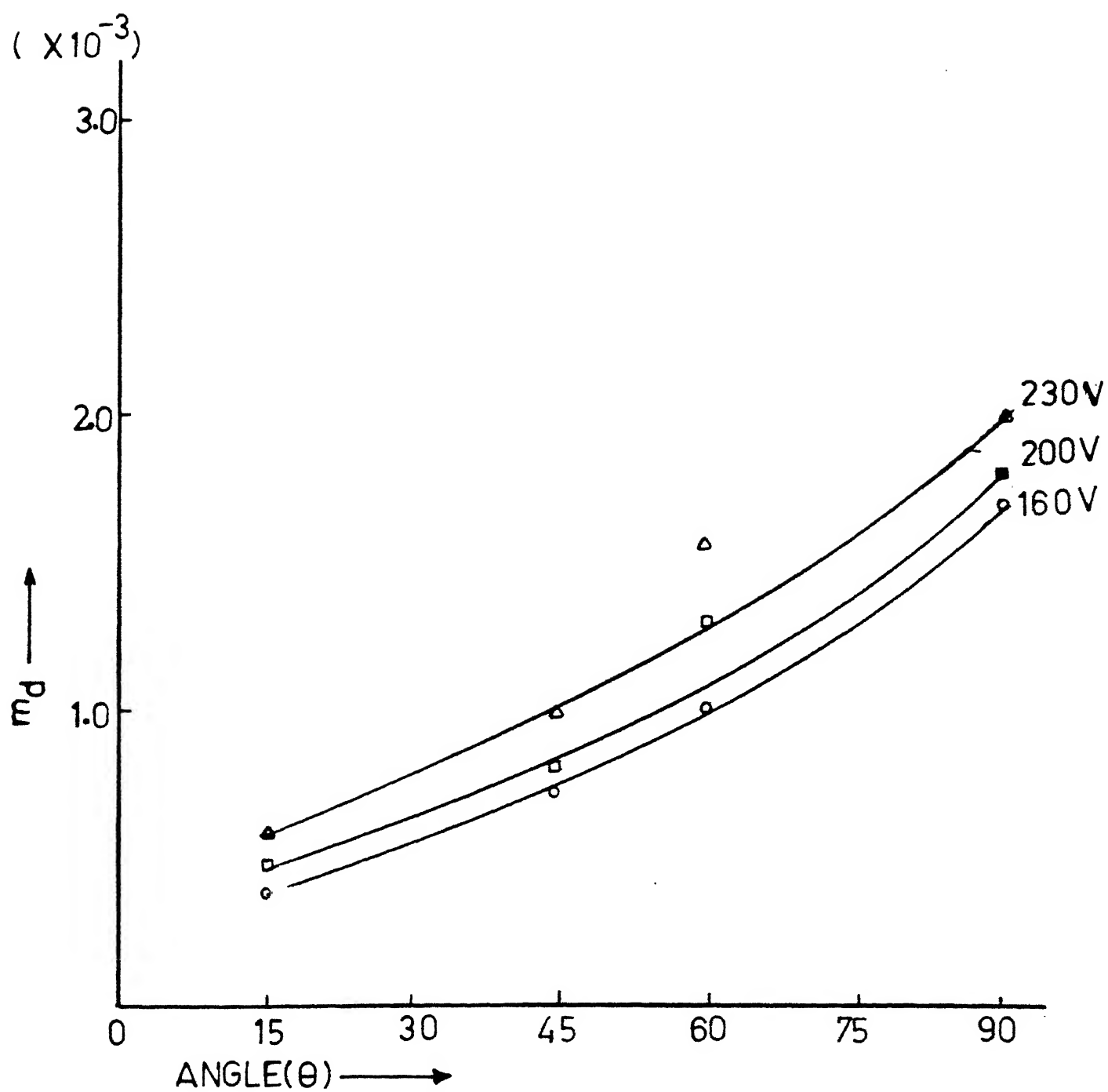0    15    30    45    60    75    90

ANGLE($\theta$) ⟶

Fig. 3.1

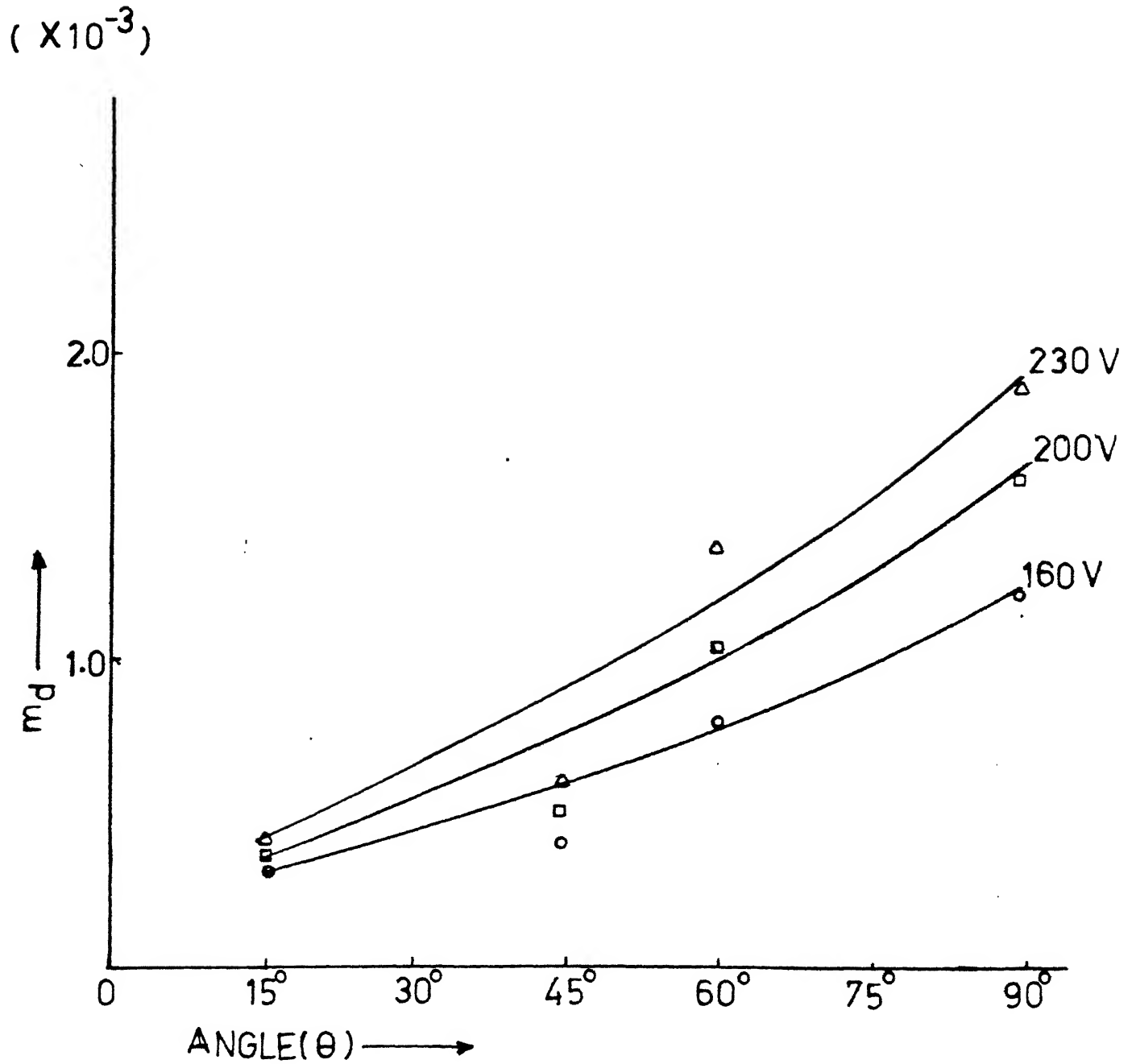Drift loss vs Inclination angle for ID fan with single stage CDE

Fig. 3.2

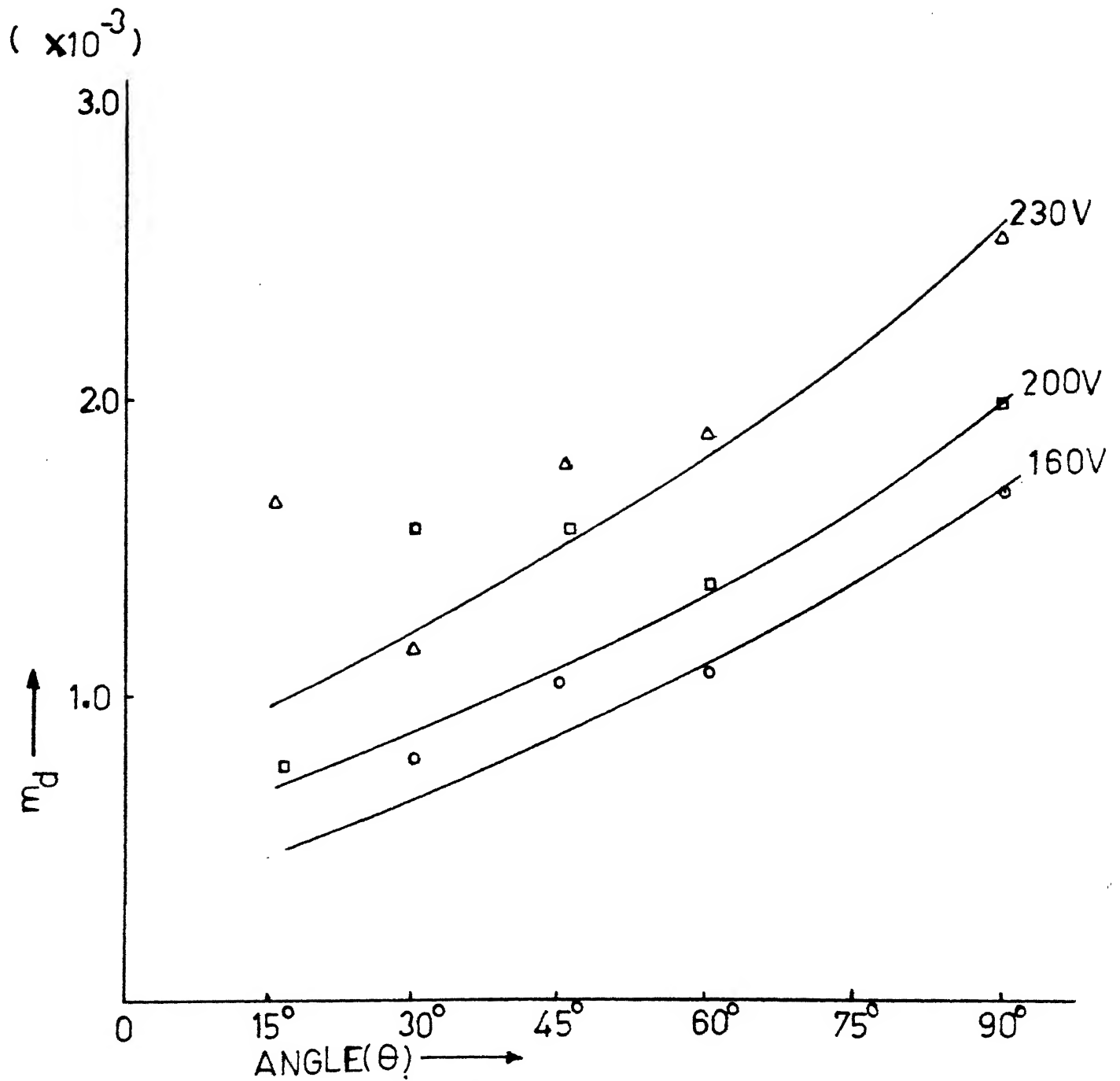Drift loss vs Inclination angle for ID fan with double stage CDE

Fig. 3.3

Drift loss vs Inclination angle for ID fan with single stage WDE

Fig. 3.4

Drift loss vs Inclination angle for ID fan with double stage WDE

Fig. 3.5

Driftloss vs Inclinatination angle for FDfanwith single stageCDE
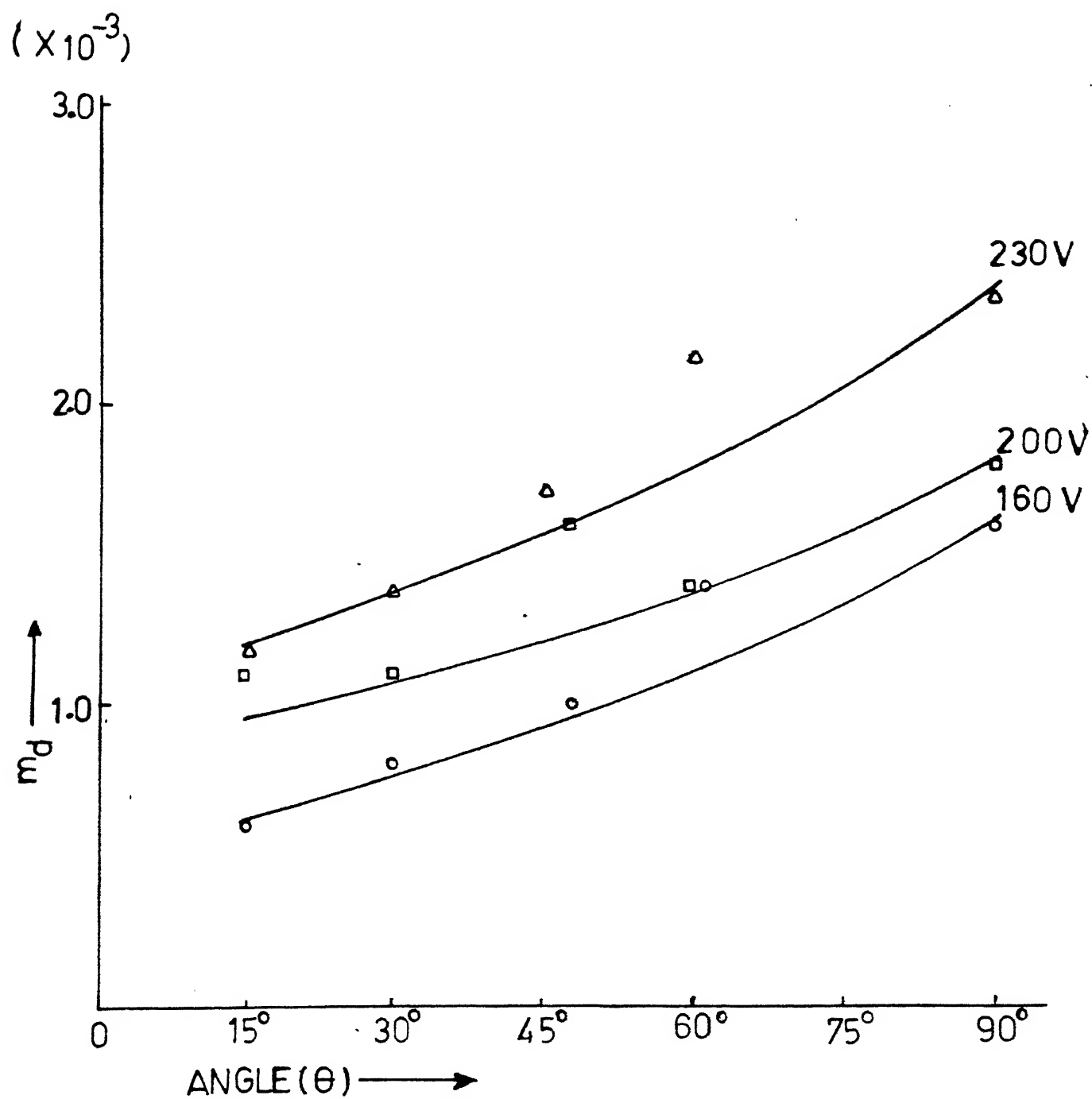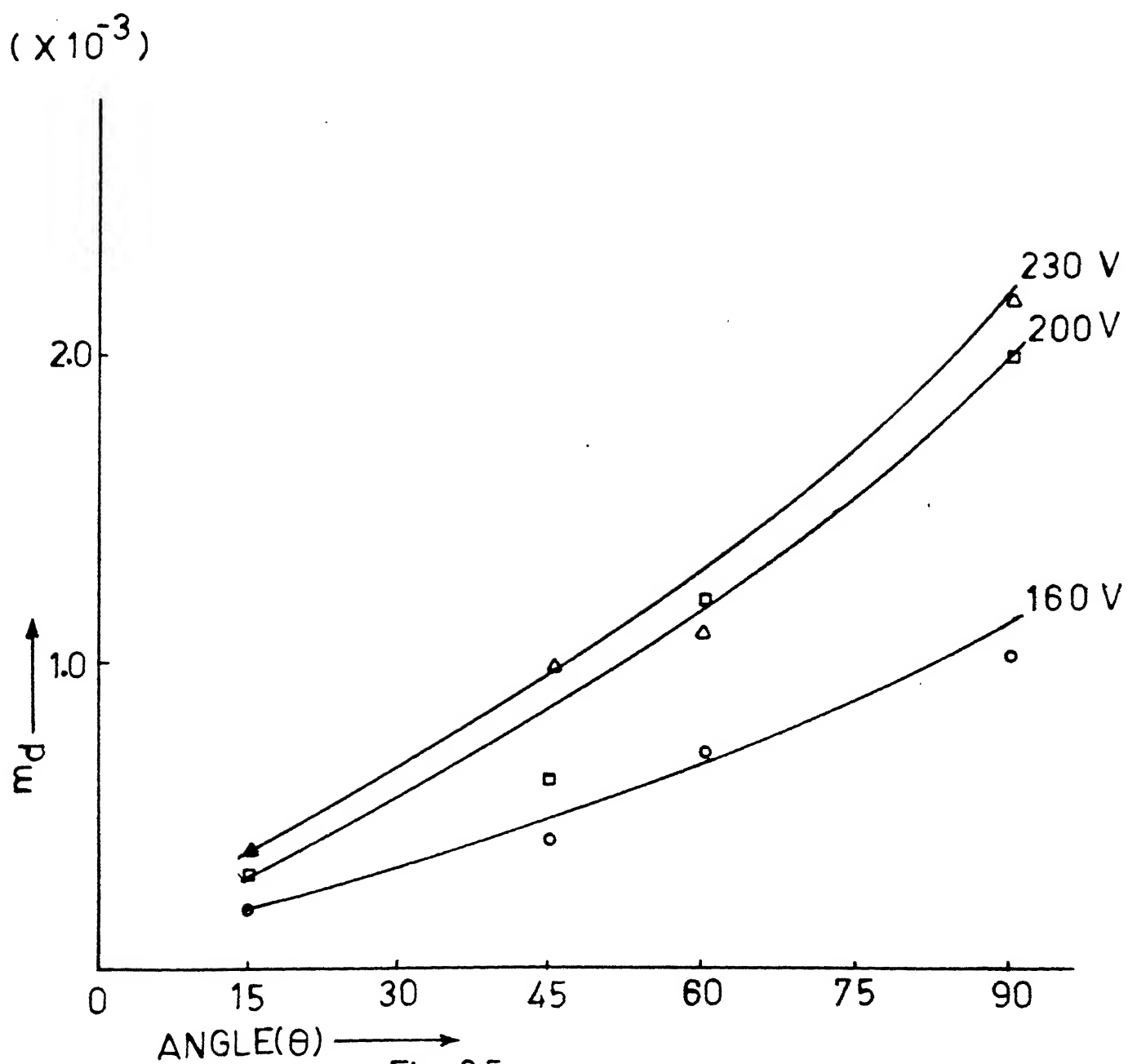
$(\times 10^{-3})$

$2.0$

$m_d \longrightarrow$

$1.0$

230 V
200 V
160 V

0    15°    30°    45°    60°    75°    90°
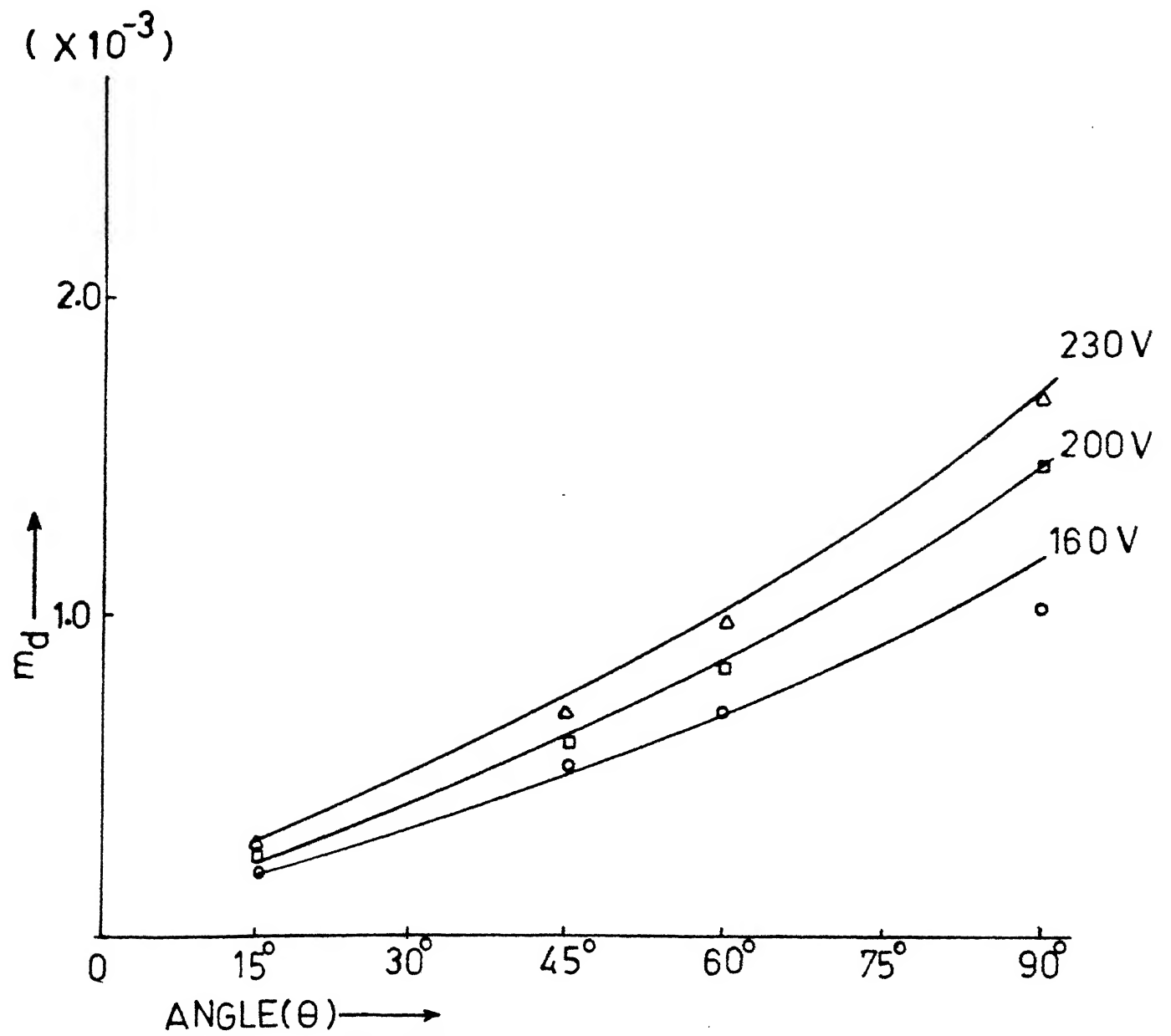
ANGLE($\theta$) $\longrightarrow$

Fig. 3.6

Drift loss vs Inclination angle for FD fan with double stage CDE

eliminators with both FD and ID fans. This difference may be attributed to the larger thickness of the concrete eliminators (25 mm) compared to that of wooden eliminators (13 mm).Air flow rates for various supply voltages are given in appendix.

3.2     PRESSURE DROP:

The pressure drop data were recorded for θ varying between 15° and 90° and for each value of θ, the supply voltage for both ID and FD fan. was varied in the range of 160 to 230 Volts A.C. in order to change the speed of the fan and hence the air discharge rate. Single and then double stages for both wooden and concrete drift eliminators were employed first with ID for and then with FD fan. The pressure drops across single stage and double stage were recorded along with the supply voltage and the power drawn by the fan in operation.

The data recorded are shown in Table 3.9 through 3.12. The pressure drop across the various stages of wooden as well as concrete eliminators with ID and FD fan in operation one at a time, are plotted versus θ in Figures 3.7 through 3.14. It can be seen from here that as θ increases, the pressure drop across a particular set of stages decreases with minimum value corresponding to θ = 90°. It can also be seen from Figures 3.8, 3.10, 3.12 and 3.14 that as the number of stages increase, the pressure drop increases which in turn requires a larger amount

## Table 3.9

### Pressure Drop Data for ID fan with WDE

| Inclination Angle $\theta$ (degree) | Supply Voltage $V$ (volts) | No.of stages $n$ | Power (fan) $P_{ID}$ (kW) | Static Pressure | | | Pressure drop $\triangle p$ (mm of $H_2O$) |
|---|---|---|---|---|---|---|---|
| | | | | $P_a$ (mm of $H_2O$) | $P_b$ (mm of $H_2O$) | $P_c$ (mm of $H_2O$) | |
| 15 | 160 | 1 | 0.36 | 8.509 | 9.652 | - | 1.143 |
| | | 2 | 0.35 | 7.62 | - | 9.144 | 1.524 |
| | 180 | 1 | 0.37 | 8.509 | 9.652 | - | 1.143 |
| | | 2 | 0.37 | 7.874 | - | 9.525 | 1.651 |
| | 200 | 1 | 0.40 | 8.382 | 9.652 | - | 1.27 |
| | | 2 | 0.37 | 8.128 | - | 9.652 | 1.524 |
| | 220 | 1 | 0.408 | 8.390 | 9.652 | - | 1.262 |
| | | 2 | 0.410 | 8.636 | - | 9.906 | 1.27 |
| | 230 | 1 | 0.46 | 8.636 | 10.16 | - | 1.524 |
| | | 2 | 0.425 | 7.874 | - | 9.906 | 2.032 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| 30 | 160 | 1 | 0.33 | 8.077 | 8.636 | – | 0.558 |
| | | 2 | 0.325 | 7.62 | – | 8.128 | 0.508 |
| | 180 | 1 | 0.33 | 8.00 | 8.382 | – | 0.382 |
| | | 2 | 0.33 | 8.128 | – | 8.636 | 0.508 |
| | 200 | 1 | 0.378 | 8.585 | 9.144 | – | 0.558 |
| | | 2 | 0.35 | 7.874 | – | 8.634 | 0.762 |
| | 220 | 1 | 0.38 | 8.636 | 9.144 | – | 0.762 |
| | | 2 | 0.375 | 7.874 | – | 8.636 | 0.762 |
| | 230 | 1 | 0.40 | 8.636 | 9.398 | – | 0.762 |
| | | 2 | 0.38 | 7.874 | – | 8.89 | 1.016 |
| 45 | 160 | 1 | 0.34 | 8.128 | 8.636 | – | 0.508 |
| | | 2 | 0.36 | 8.128 | – | 8.89 | 0.762 |
| | 180 | 1 | 0.360 | 8.001 | 8.509 | – | 0.508 |
| | | 2 | 0.36 | 7.874 | – | 8.432 | 0.558 |
| | 200 | 1 | 0.365 | 8.382 | 8.89 | – | 0.508 |
| | | 2 | 0.40 | 8.509 | – | 9.144 | 0.635 |

| | | | | | | |
|---|---|---|---|---|---|---|
| 220 | 1 | 0.425 | 8.128 | 8.509 | - | 0.381 |
| | 2 | 0.43 | 8.509 | - | 9.144 | 0.635 |
| 230 | 1 | 0.42 | 8.763 | 9.144 | - | 0.381 |
| | 2 | 0.45 | 8.765 | - | 9.652 | 0.889 |
| 160 | 1 | 0.35 | 8.382 | 8.636 | - | 0.254 |
| | 2 | 0.33 | 8.128 | - | 8.382 | 0.254 |
| 180 | 1 | 0.345 | 8.128 | 8.509 | - | 0.381 |
| | 2 | 0.35 | 7.874 | - | 8.382 | 0.508 |
| 200 | 1 | 0.375 | 8.636 | 8.89 | - | 0.254 |
| | 2 | 0.36 | 8.128 | - | 8.636 | 0.504 |
| 220 | 1 | 0.37 | 8.190 | 8.509 | - | 0.319 |
| | 2 | 0.375 | 8.001 | - | 8.255 | 0.254 |
| 230 | 1 | 0.425 | 8.89 | 9.144 | - | 0.254 |
| | 2 | 0.40 | 8.636 | - | 9.017 | 0.381 |

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| Temp | | | | | | |
|---|---|---|---|---|---|---|
| 160 | 1 | 0.36 | 8.636 | 8.636 | – | 0.0 |
| | 2 | 0.35 | 8.636 | – | 8.636 | 0.0 |
| 180 | 1 | 0.375 | 8.636 | 8.636 | – | 0.0 |
| | 2 | 0.375 | 8.636 | – | 8.636 | 0.0 |
| 200 | 1 | 0.39 | 8.636 | 8.636 | – | 0.0 |
| | 2 | 0.39 | 8.636 | – | 8.636 | 0.0 |
| 220 | 1 | 0.42 | 8.940 | 8.89 | – | 0.020 |
| | 2 | 0.42 | 8.89 | – | 8.940 | 0.05 |
| 230 | 1 | 0.44 | 9.144 | 9.017 | – | 0.050 |
| | 2 | 0.442 | 9.144 | – | 8.89 | 0.254 |

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## Table 3.10

### Pressure Drop Data for ID fan with CDE

| Inclination Angle θ (degree) | Supply Voltage V (volts) | No.of stages n | Power (fan) $P_{ID}$ (kW) | Static Pressure | | | Pressure drop ΔP (mm of $H_2O$) |
|---|---|---|---|---|---|---|---|
| | | | | $P_a$ (mm of $H_2O$) | $P_b$ (mm of $H_2O$) | $P_c$ (mm of $H_2O$) | |
| | 160 | 1 | 0.35 | 8.255 | 8.89 | - | 0.635 |
| | | 2 | 0.35 | 8.128 | - | 9.144 | 1.016 |
| | 180 | 1 | 0.358 | 8.128 | 8.703 | - | 0.635 |
| | | 2 | 0.36 | 7.62 | - | 9.144 | 1.524 |
| 15 | 200 | 1 | 0.375 | 8.636 | 9.398 | - | 0.768 |
| | | 2 | 0.375 | 8.382 | - | 9.652 | 1.27 |
| | 220 | 1 | 0.38 | 8.128 | 9.525 | - | 1.397 |
| | | 2 | 0.38 | 8.001 | - | 9.652 | 1.651 |
| | 230 | 1 | 0.38 | 8.128 | 9.271 | - | 1.143 |
| | | 2 | 0.40 | 8.128 | - | 9.70 | 1.574 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| | 160 | 1 | 0.33 | 8.128 | 8.636 | – | 0.508 |
| | | 2 | 0.34 | 7.874 | – | 8.636 | 0.762 |
| | 180 | 1 | 0.35 | 8.255 | 8.765 | – | 0.510 |
| | | 2 | 0.35 | 8.128 | – | 8.636 | 0.508 |
| | 200 | 1 | 0.36 | 8.255 | 8.89 | – | 0.635 |
| | | 2 | 0.37 | 8.128 | – | 8.89 | 0.762 |
| 45 | 220 | 1 | 0.38 | 8.128 | 9.017 | – | 8.889 |
| | | 2 | 0.38 | 8.382 | – | 9.093 | 0.711 |
| | 230 | 1 | 0.40 | 8.128 | 9.144 | – | 0.762 |
| | | 2 | 0.405 | 8.382 | – | 9.144 | 1.016 |
| | 160 | 1 | 0.35 | 8.382 | 8.636 | – | 0.254 |
| | | 2 | 0.34 | 8.128 | – | 8.382 | 0.254 |
| 60 | 180 | 1 | 0.36 | 8.128 | 8.636 | – | 0.508 |
| | | 2 | 0.35 | 8.128 | – | 8.585 | 1.02 |
| | 200 | 1 | 0.375 | 8.128 | 8.636 | – | 0.508 |
| | | 2 | 0.375 | 8.585 | – | 8.89 | 0.3048 |

| | | | | | |
|---|---|---|---|---|---|
| 220 | 1 | 0.39 | 8.636 | 9.144 | – | 0.508 |
| | 2 | 0.39 | 8.636 | – | 9.144 | 0.685 |
| 230 | 1 | 0.41 | 8.509 | 8.89 | – | 0.635 |
| | 2 | 0.42 | 8.763 | – | 9.271 | 0.889 |
| 160 | 1 | 0.37 | 8.382 | 8.382 | – | 0.0 |
| | 2 | 0.34 | 8.382 | – | 8.255 | 0.127 |
| 180 | 1 | 0.375 | 8.636 | 8.636 | – | 0.0 |
| | 2 | 0.35 | 8.128 | – | 8.509 | 0.127 |
| 200 | 1 | 0.40 | 8.686 | 8.89 | – | 0.127 |
| | 2 | 0.36 | 8.636 | – | 8.509 | 0.127 |
| 220 | 1 | 0.41 | 9.017 | 9.144 | – | 0.127 |
| | 2 | 0.375 | 8.89 | – | 8.763 | 0.127 |
| 230 | 1 | 0.425 | 8.763 | 9.144 | – | 0.127 |
| | 2 | 0.38 | 8.89 | – | 8.763 | 0.127 |

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## Table 3.11

### Pressure Drop Data for FD Fan with WDE

| Inclination Angle $\theta$ (degree) | Supply Voltage V (volts) | No.of stages n | Power (fan) $P_{FD}$ (kw) | Static Pressure | | | Pressure drop $\Delta P$ (mm of $H_2O$) |
|---|---|---|---|---|---|---|---|
| | | | | $P_a$ (mm of $H_2O$) | $P_b$ (mm of $H_2O$) | $P_c$ (mm of $H_2O$) | |
| 15 | 160 | 1 | 0.44 | 15.748 | 13.325 | – | 2.54 |
| | | 2 | 0.44 | 15.748 | – | 13.208 | 2.60 |
| | 180 | 1 | 0.48 | 16.256 | 13.716 | – | 2.54 |
| | | 2 | 0.465 | 16.256 | – | 13.716 | 2.54 |
| | 200 | 1 | 0.482 | 16.891 | 14.097 | – | 2.794 |
| | | 2 | 0.483 | 16.891 | – | 13.843 | 3.048 |
| | 220 | 1 | 0.475 | 16.764 | 13.462 | – | 3.302 |
| | | 2 | 0.476 | 16.764 | – | 13.462 | 3.302 |
| | 230 | 1 | 0.49 | 16.256 | 13.716 | – | 2.667 |
| | | 2 | 0.50 | 16.51 | – | 13.462 | 3.048 |

| | | | | | | |
|---|---|---|---|---|---|---|
| | 160 | 1 | 0.48 | 15.494 | 14.224 | – | 1.016 |
| | | 2 | 0.45 | 15.748 | – | 14.732 | 1.270 |
| | 180 | 1 | 0.492 | 16.002 | 14.732 | – | 1.27 |
| | | 2 | 0.45 | 15.748 | – | 14.732 | 1.016 |
| 30 | 200 | 1 | 0.495 | 16.256 | 15.24 | – | 1.16 |
| | | 2 | 0.45 | 16.256 | – | 14.732 | 1.524 |
| | 220 | 1 | 0.498 | 16.256 | 14.732 | – | 1.524 |
| | | 2 | 0.50 | 17.018 | – | 15.748 | 1.27 |
| | 230 | 1 | 0.51 | 16.256 | 13.716 | – | 2.54 |
| | | 2 | 0.525 | 17.018 | – | 14.224 | 2.794 |
| | 160 | 1 | 0.492 | 15.24 | 14.732 | – | 0.508 |
| | | 2 | 0.45 | 15.24 | – | 14.605 | 0.635 |
| 45 | 180 | 1 | 0.490 | 15.494 | 14.732 | – | 0.762 |
| | | 2 | 0.46 | 16.256 | – | 15.494 | 0.762 |
| | 200 | 1 | 0.492 | 15.24 | 14.478 | – | 0.762 |
| | | 2 | 0.475 | 16.51 | – | 15.494 | 1.016 |

| | | | | | |
|---|---|---|---|---|---|
| 220 | 1 | 0.495 | 15.748 | 15.24 | — | 0.508 |
| | 2 | 0.495 | 15.875 | — | 15.24 | 0.635 |
| 230 | 1 | 0.495 | 17.272 | 16.290 | — | 1.016 |
| | 2 | 0.52 | 17.018 | — | 15.748 | 1.27 |
| 160 | 1 | 0.485 | 14.985 | 14.605 | — | 0.381 |
| | 2 | 0.492 | 16.002 | — | 15.494 | 0.508 |
| 180 | 1 | 0.485 | 15.494 | 14.732 | — | 0.762 |
| | 2 | 0.495 | 16.256 | — | 15.748 | 0.508 |
| 200 | 1 | 0.486 | 14.986 | 14.859 | — | 0.127 |
| | 2 | 0.496 | 16.891 | — | 16.256 | 0.635 |
| 220 | 1 | 0.492 | 14.732 | 14.351 | — | 0.381 |
| | 2 | 0.52 | 14.732 | — | 14.224 | 0.762 |
| 230 | 1 | 0.50 | 15.24 | 14.732 | — | 0.508 |
| | 2 | 0.525 | 17.272 | — | 16.51 | 0.762 |

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| | | | | | | |
|---|---|---|---|---|---|---|
| 160 | 1 | 0.45 | 8.636 | — | 8.636 | 0.0 |
| | 2 | 0.45 | 8.636 | — | 8.636 | 0.0 |
| 180 | 1 | 0.46 | 8.636 | — | 8.636 | 0.0 |
| | 2 | 0.46 | 8.636 | — | 8.636 | 0.0 |
| 200 | 1 | 0.47 | 8.636 | — | 8.636 | 0.0 |
| | 2 | 0.47 | 8.636 | — | 8.636 | 0.0 |
| 220 | 1 | 0.47 | 8.636 | — | 8.636 | 0.0 |
| | 2 | 0.47 | 8.639 | — | 8.639 | 0.0 |
| 230 | 1 | 0.48 | 9.144 | — | 9.017 | 0.127 |
| | 2 | 0.48 | 9.144 | — | 9.017 | 0.127 |

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## Table 3.12

### Pressure Drop Data for FD fan with CDE

| Inclination Angle θ (degree) | Supply Voltage V (volts) | No. of stages n | Power (fan) $P_{FD}$ (kW) | Static Pressure | | | Pressure drop ΔP (mm of $H_2O$) |
|---|---|---|---|---|---|---|---|
| | | | | $P_a$ (mm of $H_2O$) | $P_b$ (mm of $H_2O$) | $P_c$ (mm of $H_2O$) | |
| 15 | 160 | 1 | 0.420 | 14.478 | 13.716 | - | 0.889 |
| | | 2 | 0.425 | 14.478 | - | 12.954 | 1.524 |
| | 180 | 1 | 0.425 | 14.60 | 13.716 | - | 0.884 |
| | | 2 | 0.43 | 14.73 | - | 13.716 | 1.014 |
| | 200 | 1 | 0.435 | 15.24 | 14.224 | - | 1.016 |
| | | 2 | 0.44 | 15.24 | - | 13.20 | 1.778 |
| | 220 | 1 | 0.44 | 15.24 | 13.208 | - | 2.032 |
| | | 2 | 0.45 | 15.748 | - | 13.20 | 2.54 |
| | 230 | 1 | 0.470 | 15.494 | 14.224 | - | 1.27 |
| | | 2 | 0.475 | 15.748 | - | 13.716 | 2.032 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| 45 | 160 | 1 | 0.425 | 13.97 | 13.335 | - | 0.635 |
| | | 2 | 0.425 | 13.97 | - | 13.208 | 0.762 |
| | 180 | 1 | 0.43 | 14.224 | 13.335 | - | 0.889 |
| | | 2 | 0.435 | 14.732 | - | 13.97 | 0.762 |
| | 200 | 1 | 0.45 | 15.24 | 14.35 | - | 0.889 |
| | | 2 | 0.44 | 15.24 | - | 14.22 | 1.016 |
| | 220 | 1 | 0.45 | 15.494 | 14.351 | - | 1.143 |
| | | 2 | 0.46 | 15.24 | - | 14.224 | 1.016 |
| | 230 | 1 | 0.50 | 16.00 | 14.859 | - | 1.143 |
| | | 2 | 0.48 | 15.24 | - | 14.224 | 1.27 |
| 60 | 160 | 1 | 0.425 | 13.87 | 13.462 | - | 0.508 |
| | | 2 | 0.425 | 14.22 | - | 13.589 | 0.635 |
| | 180 | 1 | 0.43 | 14.351 | 13.58 | - | 0.762 |
| | | 2 | 0.435 | 15.24 | - | 14.478 | 0.762 |
| | 200 | 1 | 0.46 | 15.24 | 14.478 | - | 0.762 |
| | | 2 | 0.44 | 14.859 | - | 13.97 | 0.508 |

| | | | | | | |
|---|---|---|---|---|---|---|
| 220 | 1 | 0.45 | 14.732 | 14.22 | - | 0.508 |
| | 2 | 0.46 | 15.24 | - | 14.478 | 0.762 |
| 230 | 1 | 0.46 | 14.732 | 15.29 | - | 0.508 |
| | 2 | 0.46 | 14.732 | - | 13.716 | 1.016 |
| 160 | 1 | 0.43 | 15.24 | 14.986 | - | 0.254 |
| | 2 | 0.42 | 13.462 | - | 13.208 | 0.254 |
| 180 | 1 | 0.42 | 15.240 | 14.986 | - | 0.259 |
| | 2 | 0.425 | 14.224 | - | 13.97 | 0.254 |
| 200 | 1 | 0.45 | 14.986 | 14.732 | - | 0.254 |
| | 2 | 0.44 | 14.478 | - | 14.224 | 0.254 |
| 220 | 1 | 0.44 | 14.448 | 14.224 | - | 0.254 |
| | 2 | 0.44 | 14.224 | - | 14.352 | 0.127 |
| 230 | 1 | 0.475 | 14.605 | 14.351 | - | 0.127 |
| | 2 | 0.46 | 14.732 | - | 14.605 | 0.127 |

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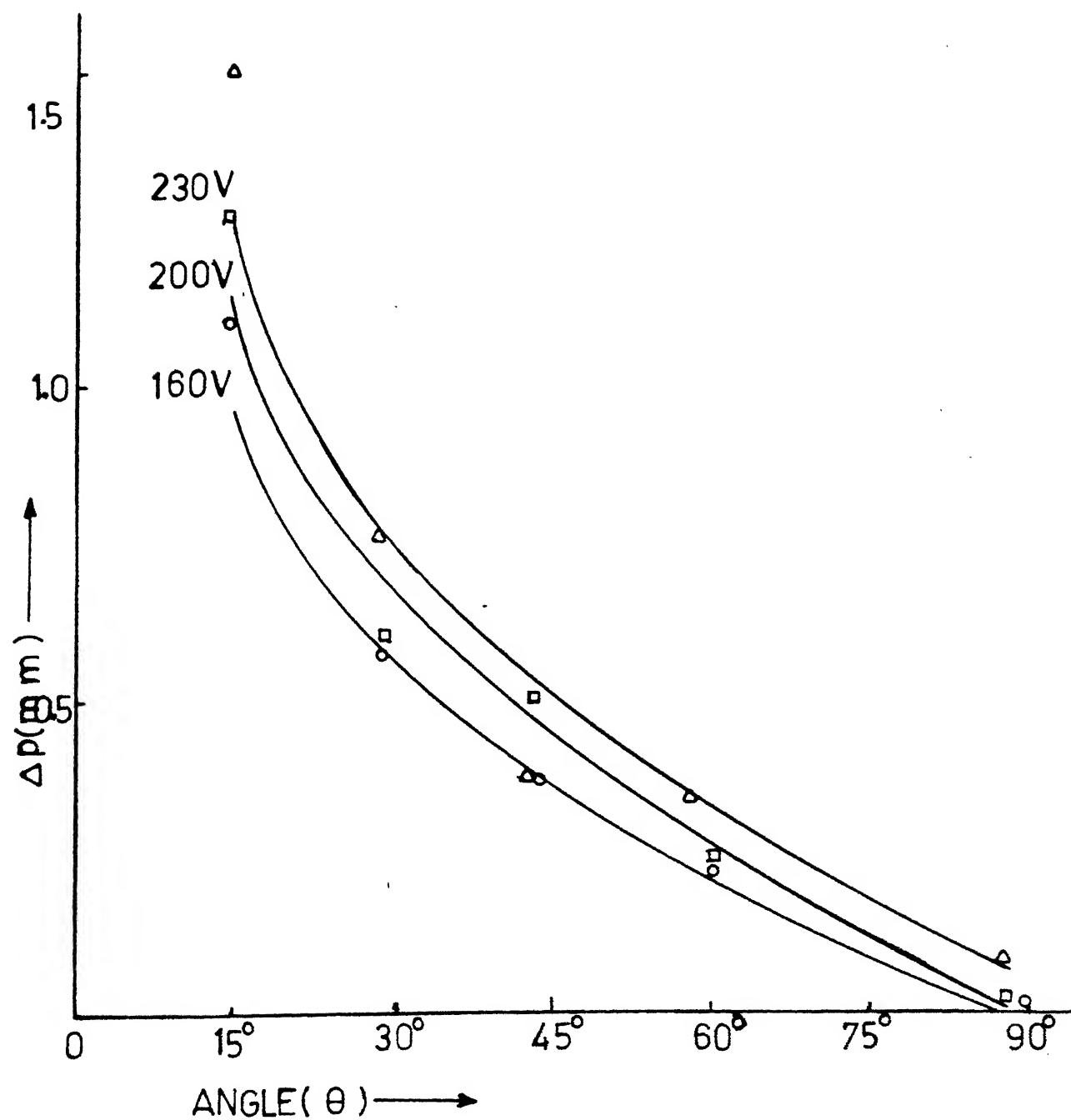
**Fig. 3.7**

Pressure drop vs inclination angle for ID fan with single stage WDE
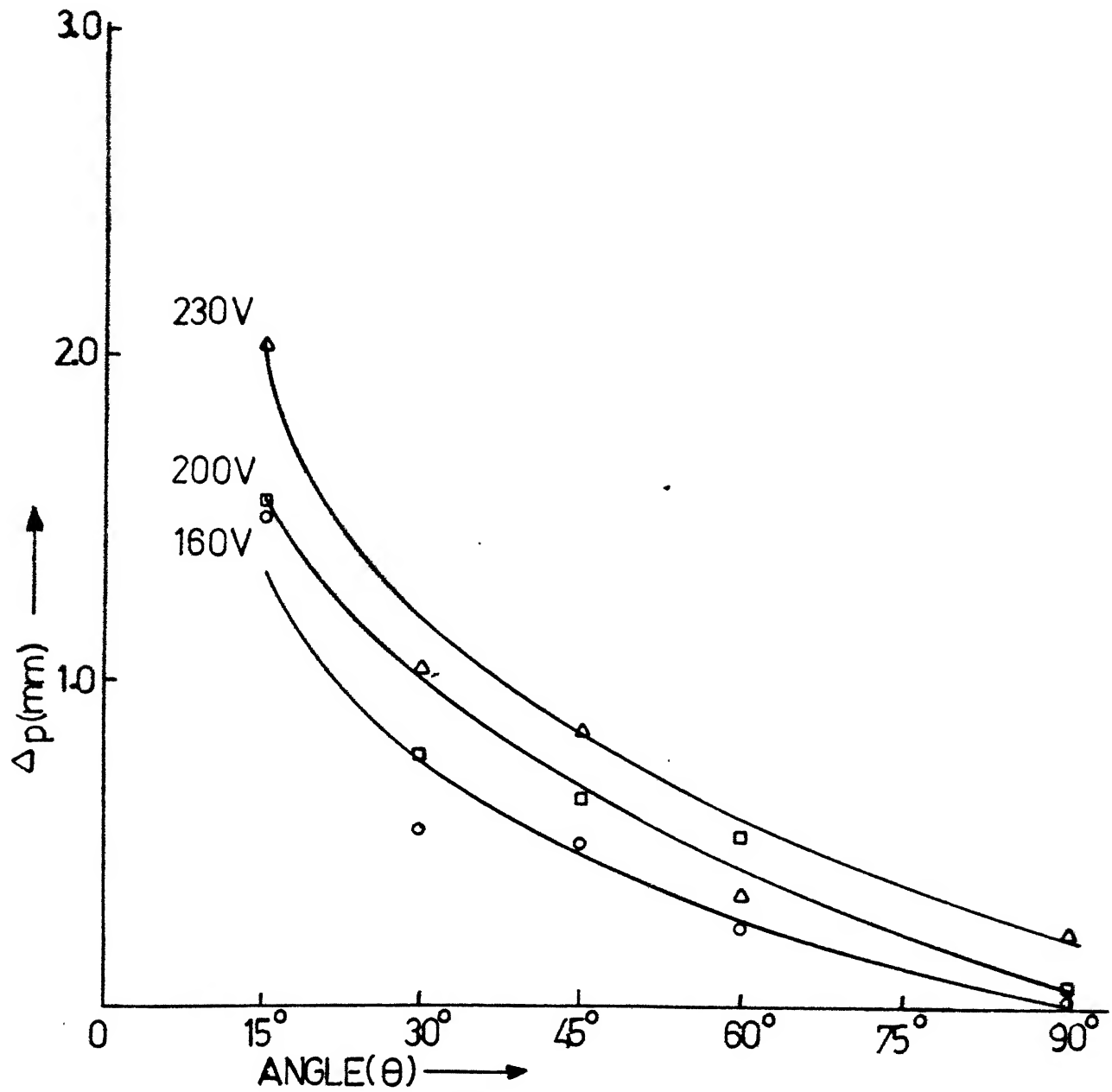
Fig.3.8

Pressure drop vs Inclination angle for ID fan with double stage WDE
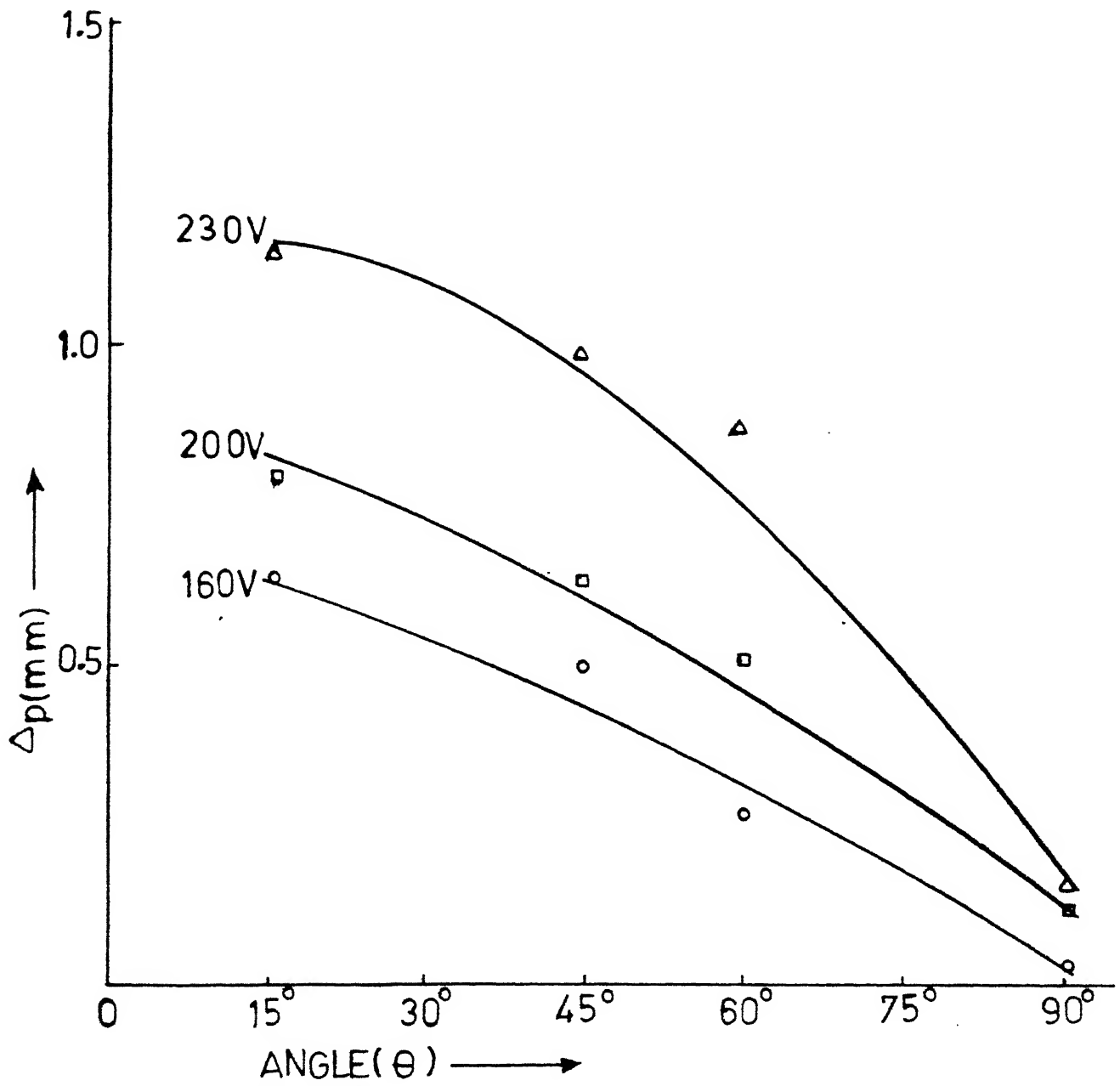
**Fig. 3.9**

Pressure drop vs Inclination angle for ID fan with single stage CDE
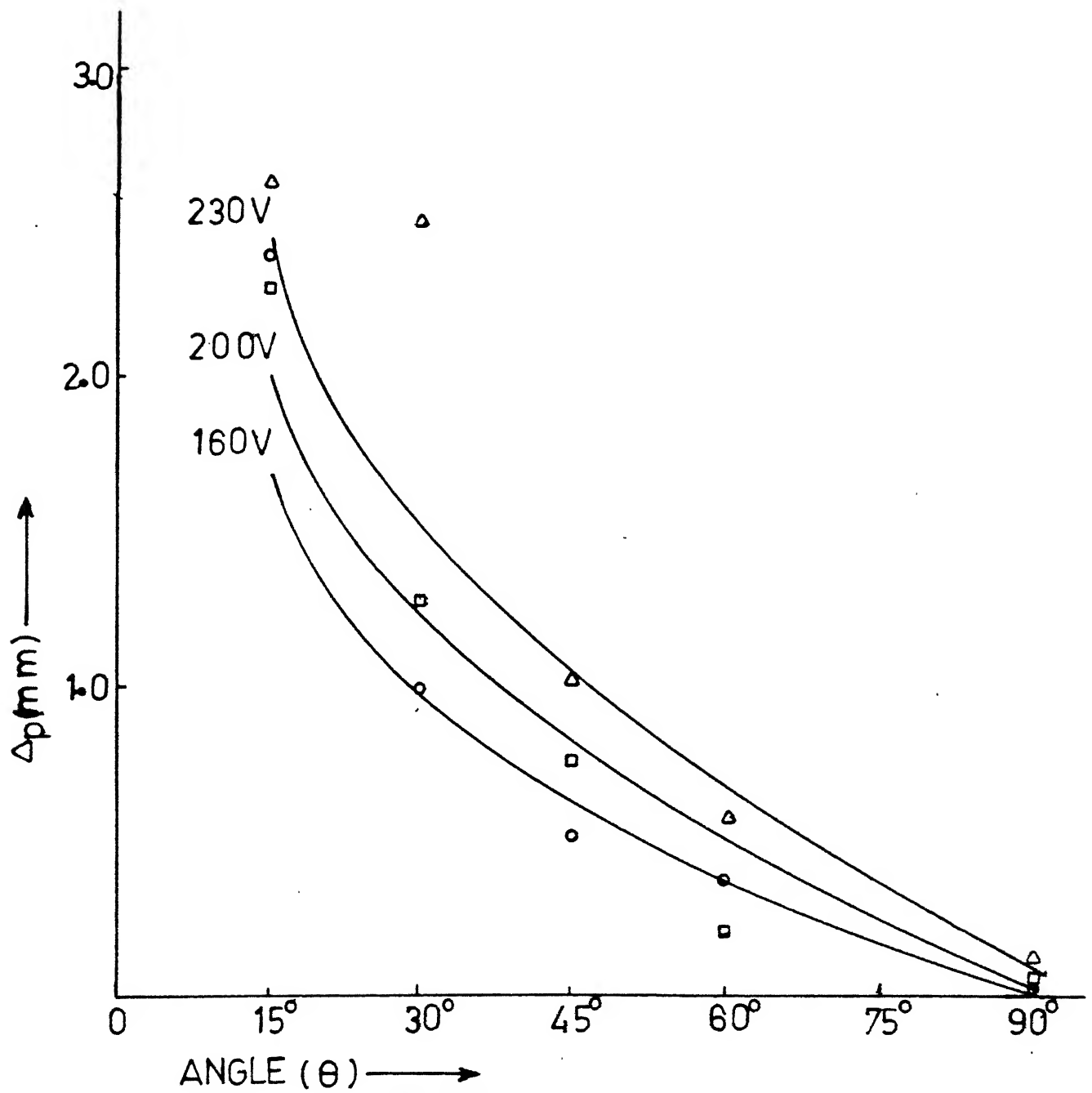
Fig. 3.11

Pressure drop vs Inclination angle for FD fan with single stage WDE
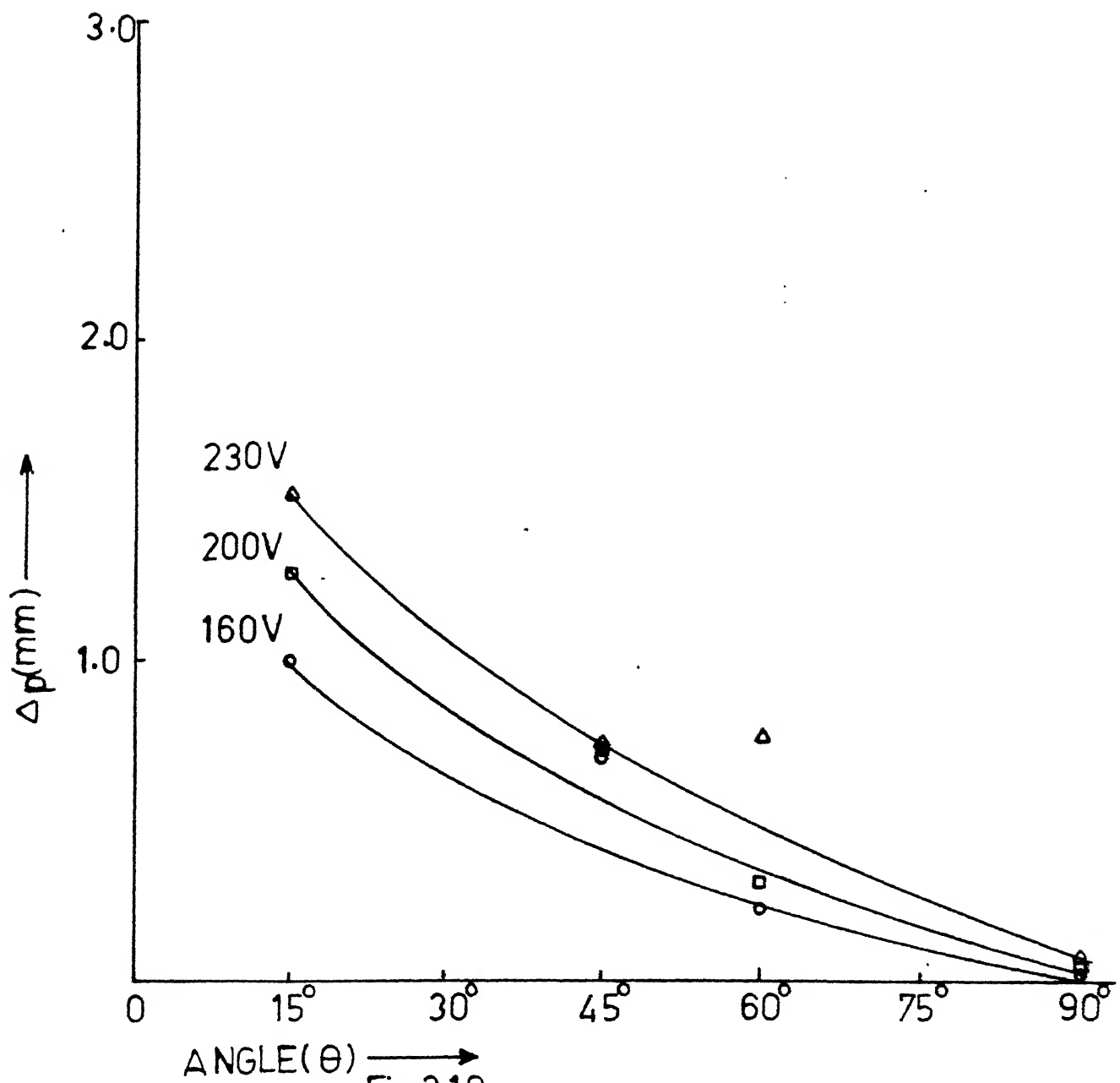
Fig. 3.10

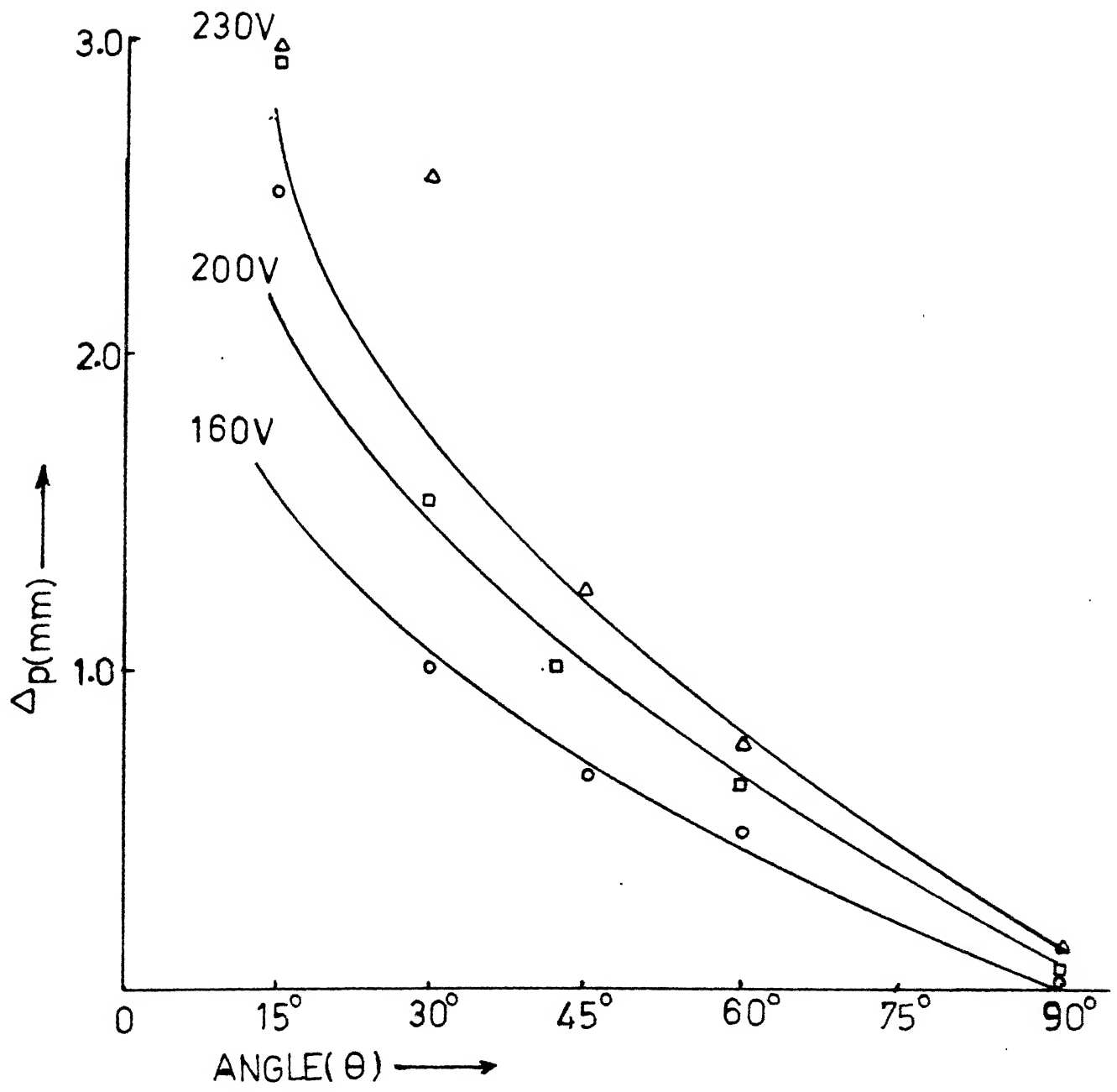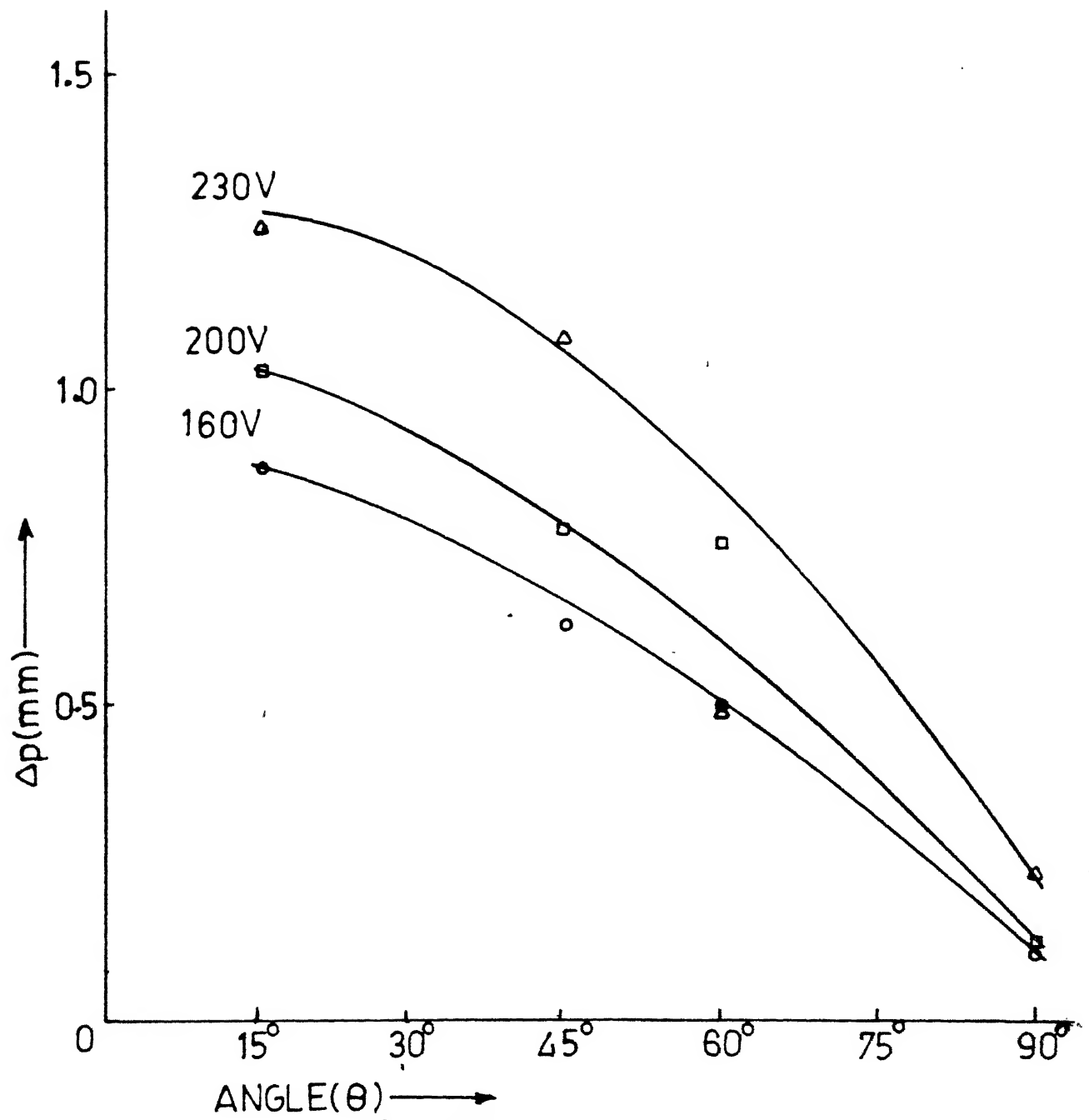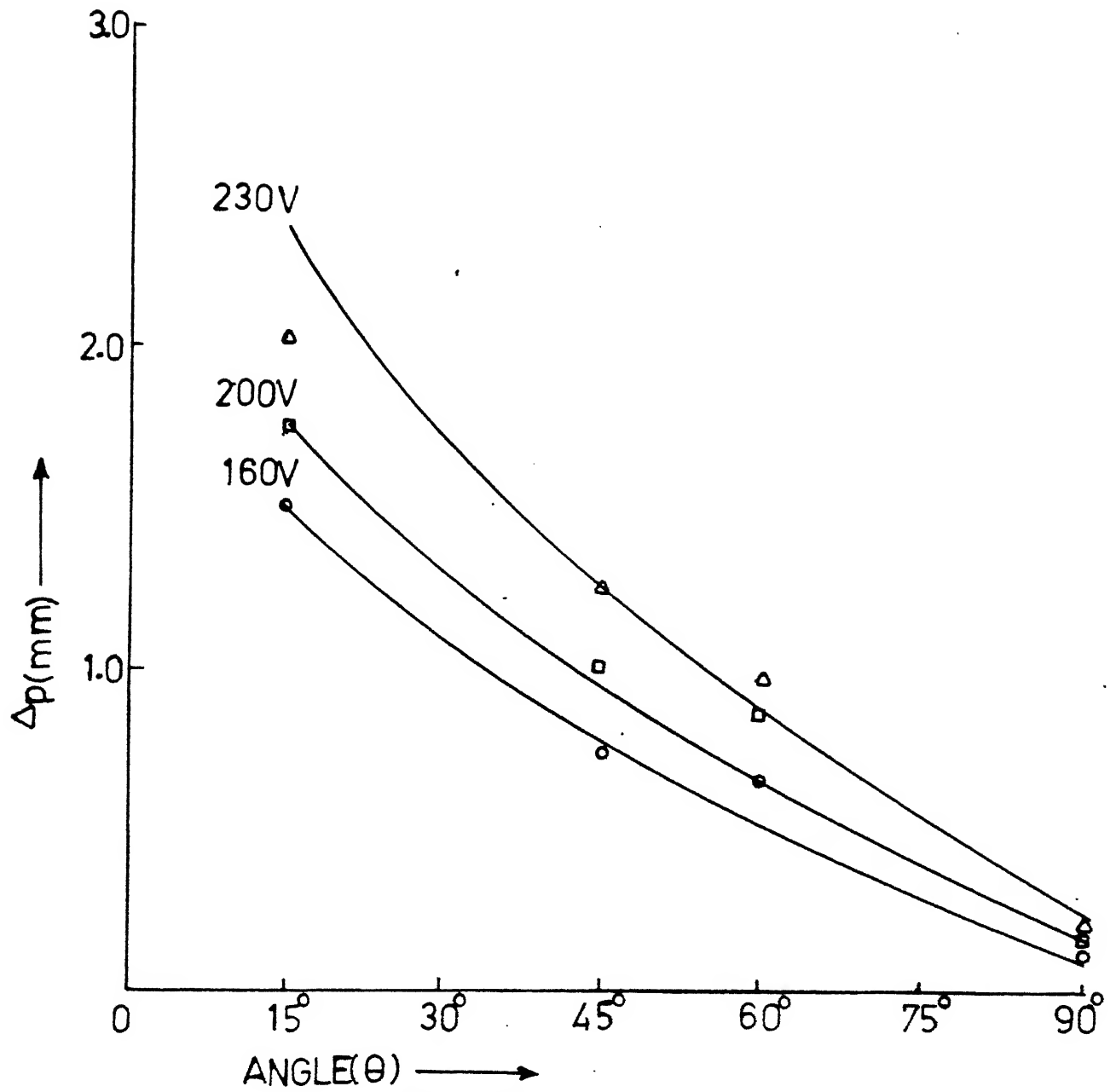Pressure drop vs Inclination angle for ID fan with double stage CDE

Fig. 3.12

Pressure drop vs Inclination angle for FD fan with double stage WDE

Fig. 3.13

Pressure drop vs Inclination angle for FD fan with single stage CDE

Fig. 3.14

Pressure drop vs Inclination angle for FD fan with double stage CDE

of power input to the fan. It is also noticed that pressure drop is more with wooden eliminators than with concrete drift eliminators. Also the pressure drop across any set of drift eliminators was more with FD fan as compared to ID fan. This may be attributed to significant amount of leakage of air which was inherent in the test rig because of improper sealing of the joints of the ducts resulting in a smaller amount of flow rate through the drift eliminators.

## 3.3    COP OF THE REFRIGERATION UNIT:

The system performance data are recorded in Tables 3.13 through 3.16 and COP is determined for various cases. The COP is plotted versus $\theta$ in Figures 3.15 through 3.22. It can be seen from here that the trend is similar for all the cases and also as the value of $\theta$ increases from $15^\circ$ to $90^\circ$, the COP for the system also increases. This is because as the $\theta$ increases the volumetric discharge of air through the system increases which results in higher evaporation rate of water thus causing more cooling of water coming in contact with evaporative condenser, thus causing subcooling of the refrigerant and bringing down the condenser pressure. From the figures it is also evident that for lower fan RPM (lower supply voltage) the COP of the system goes down, because of lower air discharge

Table 3.13

Refrigeration System Performance Data for FD fan with WDE

| Voltage V (volts) | Angle of Inclination θ (degree) | No. of stages n | Suction Pressure $P_s$ (kg/cm$^2$) | Discharge Pressure $P_d$ (kg/cm$^2$) | Condenser Inlet Temp. $T_1$ (°C) | Condenser Outlet Temp. $T_2$ (°C) | Power Input Compressor $P_{com}$ (Watts) | COP |
|---|---|---|---|---|---|---|---|---|
| 230 | 15 | 1 | 3.661 | 15.845 | 68.58 | 32.83 | 1500 | 3.876 |
| | | 2 | 3.732 | 15.774 | 69.0 | 33.2 | 1600 | 3.890 |
| | 30 | 1 | 3.521 | 15.492 | 65.83 | 29.80 | 1550 | 4.059 |
| | | 2 | 3.661 | 15.492 | 96.08 | 32.28 | 1730 | 2.452 |
| | 45 | 1 | 3.802 | 15.915 | 90.30 | 33.10 | 1580 | 2.673 |
| | | 2 | 3.345 | 15.492 | 68.0 | 29.53 | 1540 | 3.830 |
| | 60 | 1 | 3.873 | 16.126 | 86.73 | 33.65 | 1595 | 2.866 |
| | | 2 | 4.049 | 15.845 | 24.43 | 32.83 | 1780 | 2.547 |
| | 90 | 1 | 3.500 | 15.674 | 64.80 | 29.80 | 1540 | 4.115 |
| | | 2 | 3.502 | 15.774 | 64.90 | 30.00 | 1500 | 4.205 |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| 15 | | 1 | 3.732 | 16.197 | 80.83 | 32.83 | 1590 | 3.070 |
| | | 2 | 3.767 | 16.338 | 80.5 | 33.20 | 1600 | 3.160 |
| 30 | | 1 | 3.697 | 15.633 | 78.48 | 30.0 | 1575 | 3.204 |
| | | 2 | 4.225 | 16.408 | 97.93 | 30.0 | 1700 | 2.601 |
| 45 | 200 | 1 | 3.943 | 16.056 | 92.23 | 33.65 | 1580 | 2.819 |
| | | 2 | 3.6619 | 15.492 | 84.53 | 31.73 | 1640 | 2.950 |
| 60 | | 1 | 3.877 | 15.774 | 88.93 | 33.65 | 1595 | 2.807 |
| | | 2 | 3.943 | 15.845 | 91.68 | 32.83 | 1690 | 2.650 |
| 90 | | 1 | 3.697 | 15.640 | 76.48 | 31.00 | 1580 | 3.325 |
| | | 2 | 3.710 | 15.660 | 78.49 | 30.39 | 1670 | 3.350 |

| | | | | | | |
|---|---|---|---|---|---|---|
| 15 | 1 | 4.014 | 16.197 | 83.43 | 32.83 | 1625 | 2.949 |
| | 2 | 4.084 | 16.338 | 86.20 | 33.3 | 1650 | 2.909 |
| 30 | 1 | 3.873 | 15.845 | 86.18 | 29.80 | 1550 | 2.867 |
| | 2 | 3.732 | 16.197 | 81.23 | 30.0 | 1700 | 3.09 |
| 45 | 1 | 3.873 | 15.845 | 90.03 | 33.10 | 1550 | 2.688 |
| | 2 | 3.521 | 15.492 | 78.48 | 30.35 | 1590 | 3.166 |
| 60 | 1 | 3.873 | 16.056 | 94.43 | 33.10 | 1590 | 2.506 |
| | 2 | 4.014 | 15.492 | 89.48 | 33.38 | 1650 | 2.736 |
| 90 | 11 | 3.873 | 15.492 | 78.48 | 30.35 | 1580 | 3.202 |
| | 2 | 3.875 | 15.495 | 73.60 | 30.50 | 1590 | 3.200 |

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## Table 3.14

### Refrigeration System Performance Data for FD fan with CDE

| Voltage V (Volts) | Angle of Inclination θ (degree) | No. of stages n | Suction Pressure $P_s$ (kg/cm$^2$) | Discharge Pressure $P_d$ (kg/cm$^2$) | Condenser Inlet Temp. $T_1$ (°C) | Condenser Outlet Temp. $T_2$ (°C) | Power Input Compressor $P_{com}$ (Watts) | COP |
|---|---|---|---|---|---|---|---|---|
| 230 | 15 | 1 | 4.084 | 16.549 | 97.18 | 29.53 | 1660 | 2.478 |
|  |  | 2 | 4.30 | 16.971 | 99.38 | 33.38 | 1610 | 2.414 |
|  | 45 | 1 | 4.225 | 16.690 | 101.58 | 32.28 | 1750 | 2.342 |
|  |  | 2 | 4.295 | 16.971 | 97.18 | 32.83 | 1640 | 2.486 |
|  | 60 | 1 | 4.295 | 16.760 | 106.53 | 32.83 | 1750. | 2.222 |
|  |  | 2 | 4.366 | 16.901 | 92.78 | 32.28 | 1610 | 2.700 |
|  | 90 | 1 | 4.084 | 16.549 | 93.88 | 30.63 | 1600 | 2.617 |
|  |  | 2 | 3.767 | 16.197 | 82.33 | 29.80 | 1550 | 3.053 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| 200 | 15 | 1 | 4.084 | 16.760 | 101.03 | 29.8 | 1650 | 1.962 |
| | | 2 | 4.225 | 16.971 | 103.7 | 33.10 | 1600 | 1.964 |
| | 45 | 1 | 4.295 | 16.830 | 109.8 | 31.73 | 1770 | 2.122 |
| | | 2 | 4.260 | 17.042 | 104.88 | 32.83 | 1650 | 1.965 |
| | 60 | 1 | 4.225 | 16.830 | 109.2 | 32.28 | 1720 | 2.083 |
| | | 2 | 4.295 | 17.042 | 104.33 | 32.55 | 1650 | 1.968 |
| | 90 | 1 | 4.154 | 16.760 | 102.13 | 30.90 | 1620 | 2.328 |
| | | 2 | 3.943 | 16.338 | 90.58 | 30.0 | 1560 | 2.677 |
| 160 | 15 | 1 | 4.225 | 16.549 | 105.43 | 30.63 | 1700 | 2.231 |
| | | 2 | 4.436 | 17.042 | 104.8 | 33.38 | 1650 | 2.088 |
| | 45 | 1 | 4.225 | 16.830 | 108.18 | 32.28 | 1750 | 2.134 |
| | | 2 | 4.154 | 17.042 | 106.5 | 32.83 | 1650 | 2.194 |
| | 60 | 1 | 4.366 | 16.901 | 109.83 | 31.73 | 1710 | 1.963 |
| | | 2 | 4.225 | 16.971 | 105.70 | 32.28 | 1650 | 2.230 |
| | 90 | 1 | 4.084 | 16.549 | 105.40 | 30.63 | 1600 | 2.226 |
| | | 2 | 4.049 | 16.619 | 98.28 | 30.63 | 1600 | 2.416 |

Table 3.15

Refrigeration System Performance Data for ID fan with WDE

| Voltage V (volts) | Angle of Inclination θ (degree) | No. of stages n | Suction Pressure $P_s$ (kg/cm²) | Discharge Pressure $P_d$ (kg/cm²) | Condenser Inlet Temp. $T_1$ (°C) | Condenser Outlet Temp. $T_2$ (°C) | Power Input Compressor $P_{com}$ (watts) | COP |
|---|---|---|---|---|---|---|---|---|
| 230 | 15 | 1 | 4.507 | 17.464 | 114.78 | 32.83 | 2020 | 2.010 |
| | | 2 | 4.401 | 17.464 | 111.48 | 34.75 | 1950 | 2.081 |
| | 30 | 1 | 4.084 | 17.253 | 98.83 | 28.98 | 1700 | 2.416 |
| | | 2 | 4.366 | 17.394 | 104.88 | 31.18 | 1700 | 2.262 |
| | 45 | 1 | 4.401 | 17.429 | 111.48 | 30.63 | 1780 | 2.076 |
| | | 2 | 4.401 | 17.429 | 110.93 | 34.48 | 1900 | 2.082 |
| | 60 | 1 | 4.436 | 17.429 | 110.93 | 31.73 | 1800 | 2.286 |
| | | 2 | 4.330 | 17.253 | 101.03 | 32.28 | 1700 | 2.354 |
| | 90 | 1 | 4.401 | 17.253 | 92.23 | 33.54 | 1900 | 2.701 |
| | | 2 | 4.415 | 17.254 | 92.3 | 33.54 | 1880 | 2.700 |

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| | | | | | | |
|---|---|---|---|---|---|---|
| 15 | 1 | 4.330 | 17.3943 | 116.98 | 32.55 | 1950 | 1.944 |
| | 2 | 4.225 | 17.3239 | 117.53 | 33.10 | 1915 | 1.906 |
| 30 | 1 | 4.366 | 17.394 | 103.23 | 29.53 | 1720 | 2.279 |
| | 2 | 4.401 | 17.394 | 110.93 | 30.63 | 1700 | 2.087 |
| 45 | 1 | 4.401 | 17.394 | 113.13 | 30.63 | 1730 | 2.026 |
| | 2 | 4.225 | 17.323 | 115.8 | 33.93 | 1820 | 1.954 |
| 60 | 1 | 4.225 | 17.323 | 113.13 | 31.18 | 1810 | 2.010 |
| | 2 | 4.401 | 17.253 | 107.08 | 32.0 | 1700 | 2.189 |
| 90 | 1 | 4.401 | 17.253 | 105.4 | 30.0 | 1950 | 2.356 |
| | 2 | 4.403 | 17.254 | 104.5 | 30.5 | 1940 | 2.340 |

200

| | | | | | | |
|---|---|---|---|---|---|---|
| 15 | 1 | 4.330 | 17.394 | 117.8 | 32.0 | 1985 | 1.906 |
| | 2 | 4.577 | 17.535 | 117.5 | 34.2 | 1950 | 1.931 |
| 30 | 1 | 4.401 | 17.394 | 107.0 | 29.25 | 1720 | 2.192 |
| | 2 | 4.401 | 17.394 | 112.30 | 30.63 | 1700 | 2.057 |
| 45 | 1 | 4.429 | 17.464 | 113.68 | 31.45 | 1725 | 2.032 |
| | 2 | 4.788 | 17.429 | 116.70 | 35.90 | 1800 | 2.000 |
| 60 | 1 | 4.401 | 17.429 | 113.68 | 31.18 | 1810 | 2.027 |
| | 2 | 4.366 | 17.394 | 109.2 | 31.18 | 1695 | 2.119 |
| 90 | 1 | 4.577 | 17.429 | 115.8 | 30.08 | 1990 | 2.001 |
| | 2 | 4.560 | 17.5 | 116.8 | 30.00 | 2000 | 2.040 |

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## Table 3.16

### Refrigeration System Performance Data for ID fan with CDE

| Voltage V (volts) | Angle of Inclination θ (degree) | No. of stages n | Suction Pressure $p_s$ (kg/cm²) | Discharge Pressure $p_d$ (kg/cm²) | Condenser Inlet Temp. $T_1$ (°C) | Condenser Outlet Temp. $T_2$ (°C) | Power Input Compressor $P_{com}$ (watts) | COP |
|---|---|---|---|---|---|---|---|---|
| 230 | 15 | 1 | 4.401 | 16.830 | 112.0 | 30.08 | 1680 | 2.471 |
|  | 15 | 2 | 4.154 | 16.901 | 96.63 | 30.90 | 1600 | 2.518 |
|  | 45 | 1 | 4.366 | 17.042 | 113.68 | 33.38 | 1800 | 2.052 |
|  | 45 | 2 | 4.295 | 16.971 | 109.83 | 31.73 | 1690 | 2.120 |
|  | 60 | 1 | 4.401 | 16.901 | 108.73 | 31.73 | 1800 | 2.151 |
|  | 60 | 2 | 4.154 | 16.971 | 100.5 | 30.63 | 1700 | 2.357 |
|  | 90 | 1 | 4.401 | 17.147 | 114.23 | 33.65 | 1950 | 2.037 |
|  | 90 | 2 | 4.401 | 17.253 | 105.43 | 33.93 | 1670 | 2.245 |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 15 | 1 | 4.401 | 17.147 | 114.23 | 33.65 | 1950 | 2.023 |
| | | 2 | 4.366 | 17.323 | 109.55 | 33.38 | 1660 | 2.008 |
| | 45 | 1 | 4.507 | 17.112 | 114.78 | 33.38 | 1790 | 2.035 |
| | | 2 | 4.084 | 16.901 | 111.7 | 31.73 | 1660 | 2.331 |
| 00 | 60 | 1 | 4.401 | 17.112 | 112.0 | 31.45 | 1810 | 2.067 |
| | | 2 | 4.154 | 16.901 | 108.18 | 30.35 | 1680 | 2.132 |
| | 90 | 1 | 4.084 | 16.549 | 99.93 | 30.08 | 1700 | 2.386 |
| | | 2 | 4.154 | 16.901 | 96.63 | 30.90 | 1600 | 2.510 |
| | 15 | 1 | 4.436 | 17.112 | 117.53 | 33.93 | 1950 | 1.956 |
| | | 2 | 4.401 | 17.464 | 112.03 | 33.38 | 1710 | 2.066 |
| | 45 | 1 | 4.436 | 17.077 | 115.33 | 32.28 | 1800 | 2.002 |
| | | 2 | 4.436 | 17.183 | 113.13 | 32.0 | 1700 | 2.034 |
| 160 | 60 | 1 | 4.295 | 17.042 | 115.88 | 31.73 | 1950 | 1.989 |
| | | 2 | 4.436 | 17.112 | 111.48 | 30.35 | 1700 | 2.085 |
| | 90 | 1 | 4.225 | 16.901 | 111.48 | 30.35 | 1715 | 2.115 |
| | | 2 | 4.40 | 17.112 | 109.83 | 31.18 | 1675 | 2.109 |
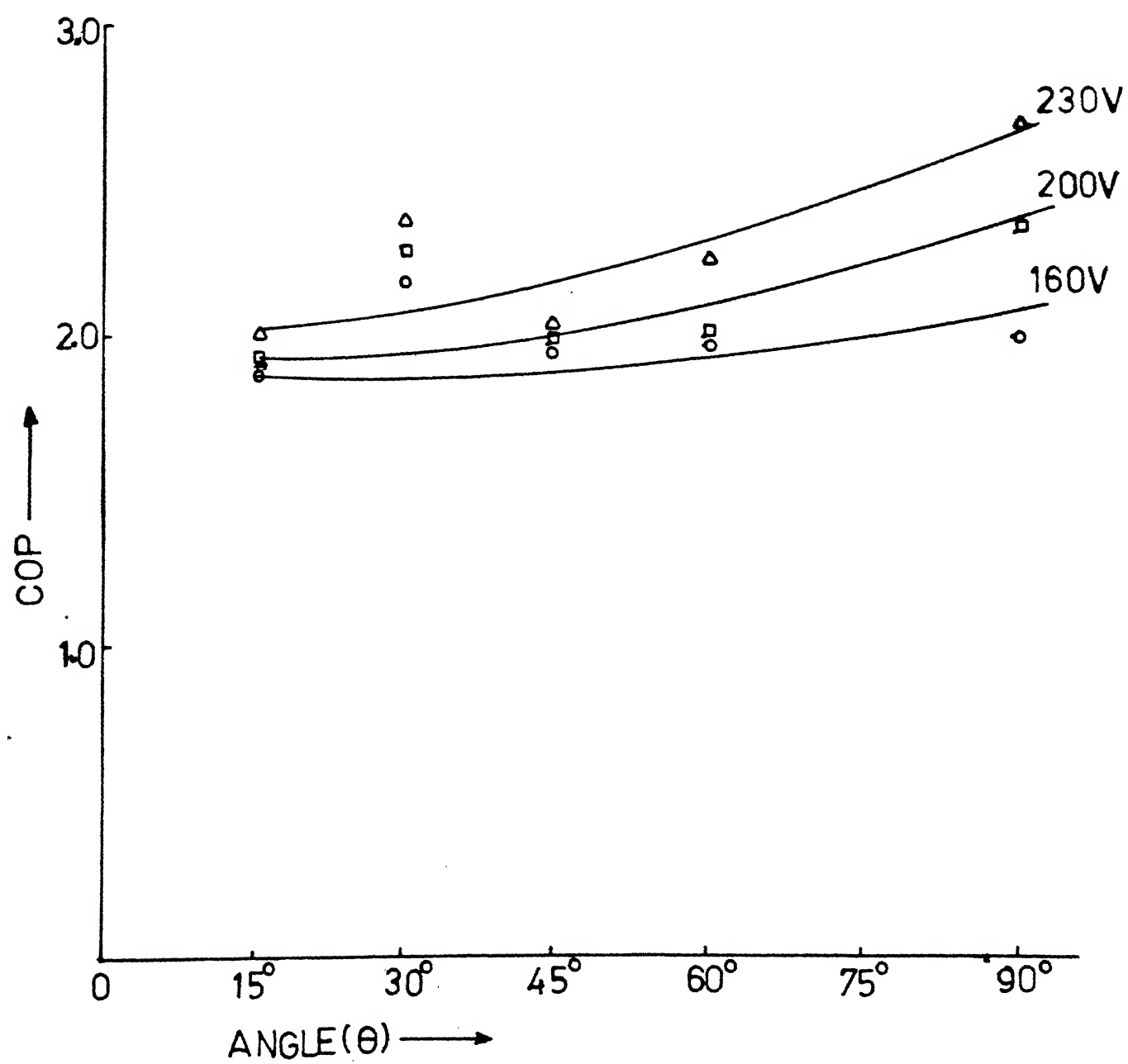
Fig. 3.15

COP vs Inclination angle for ID fan with single stage WDE
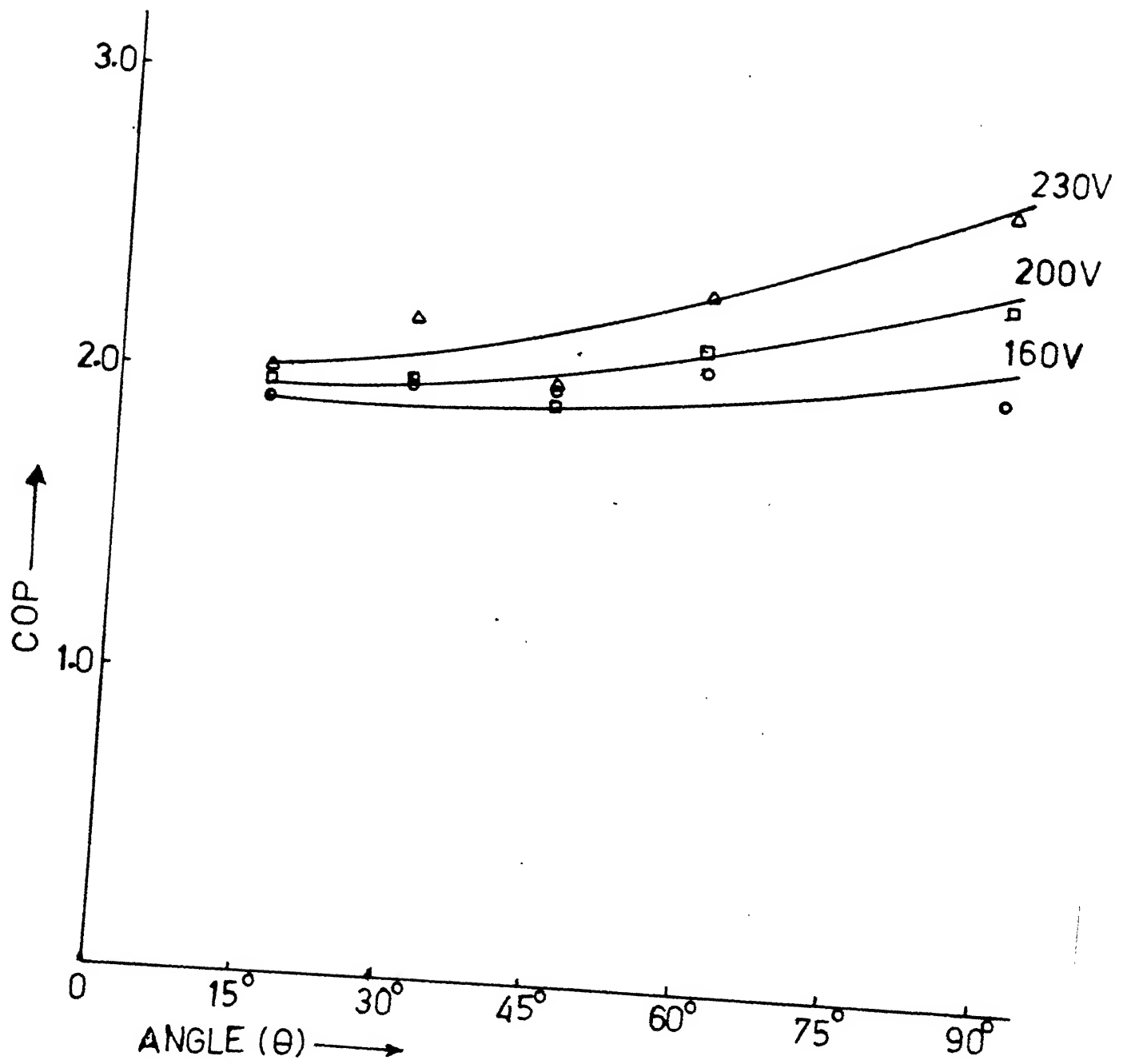
Fig. 3.16

COP vs Inclination angle for ID fan with double stage WDE
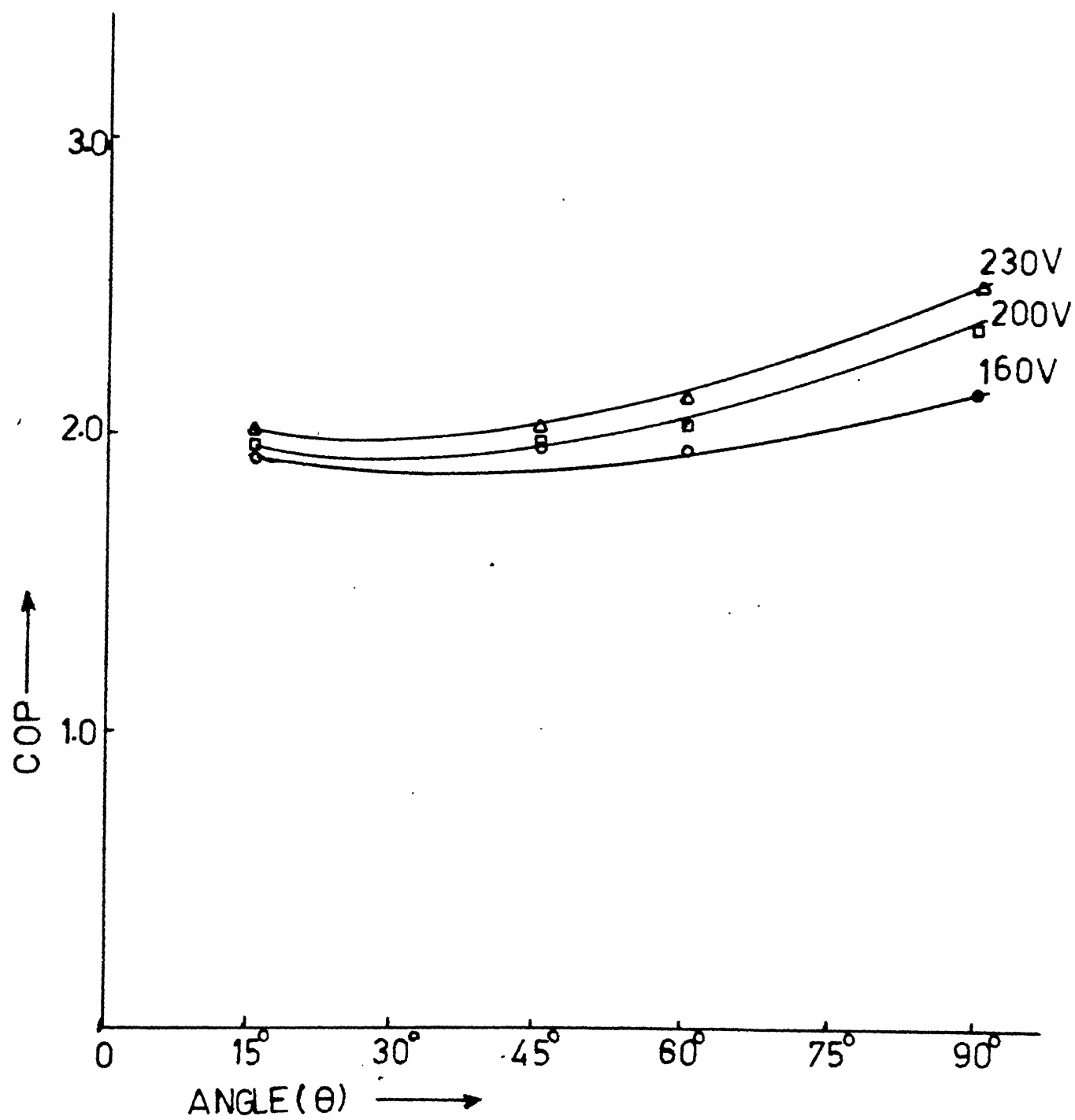
<u>Fig. 3.17</u>

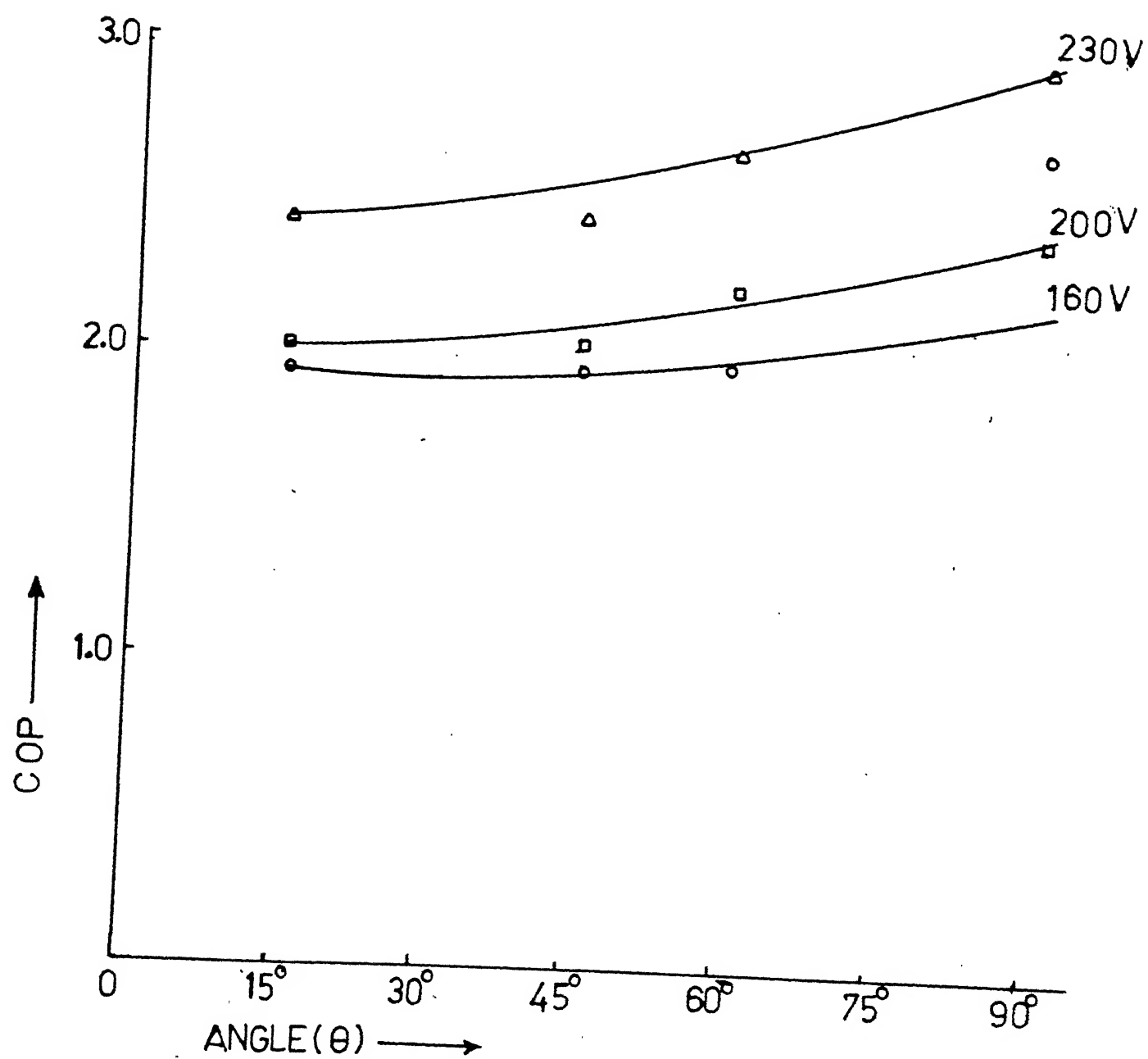COP vs Inclination angle for IDfan with single stage CDE
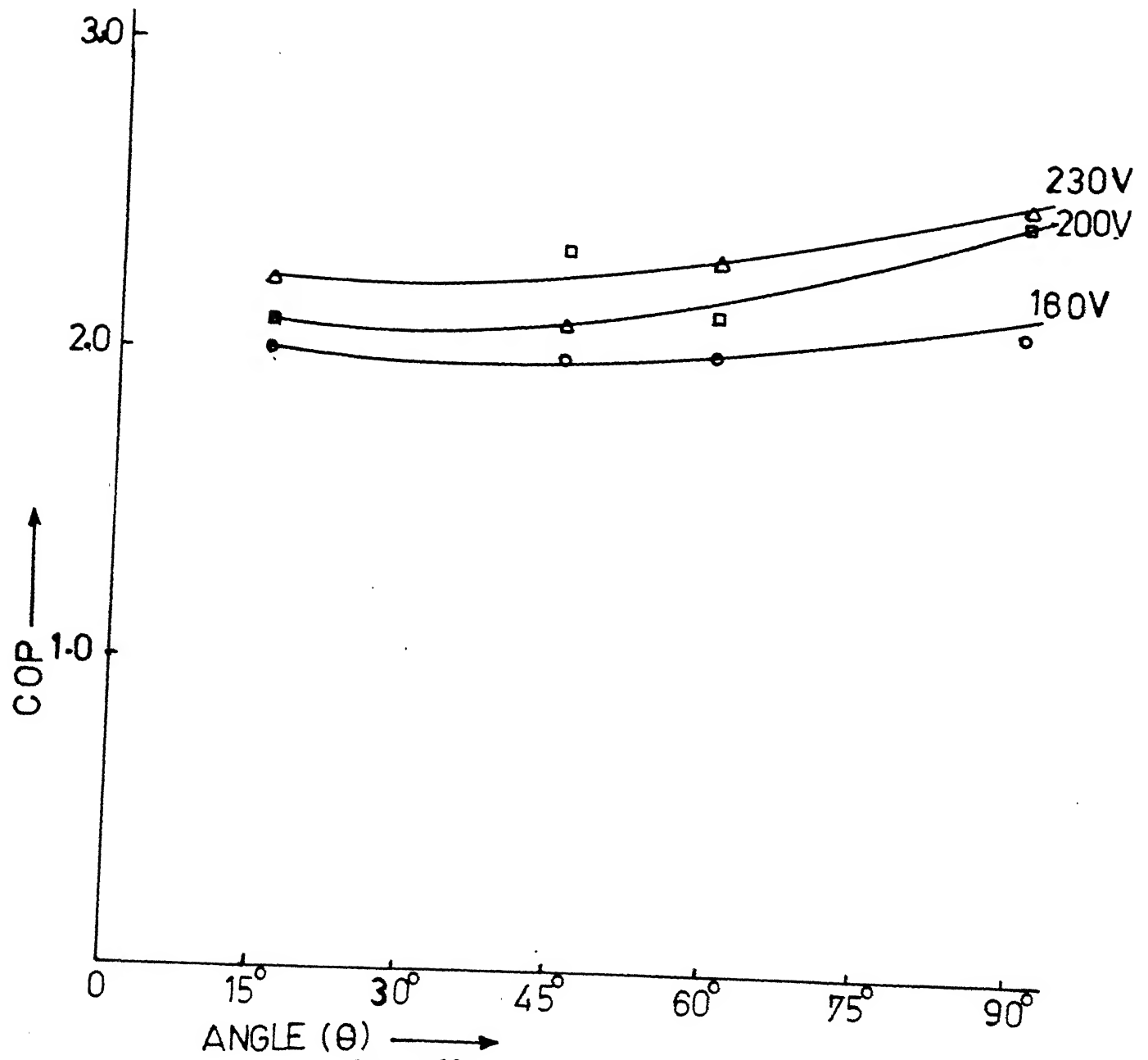
Fig. 3.18

COP vs Inclination angle for ID fan with double stage CDE
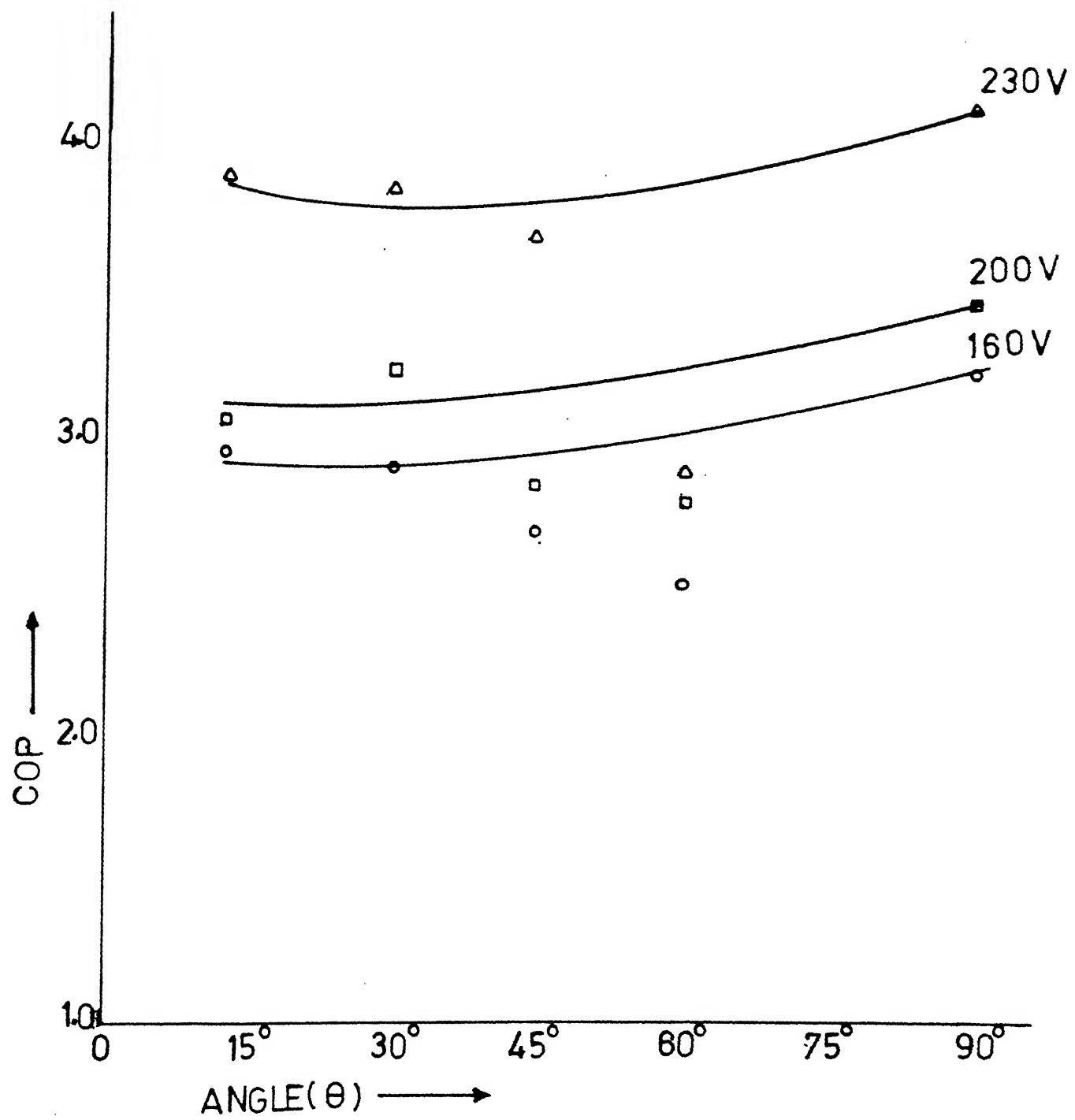
Fig. 3.19

COP vs Inclination angle for FD fan with single stage WDE
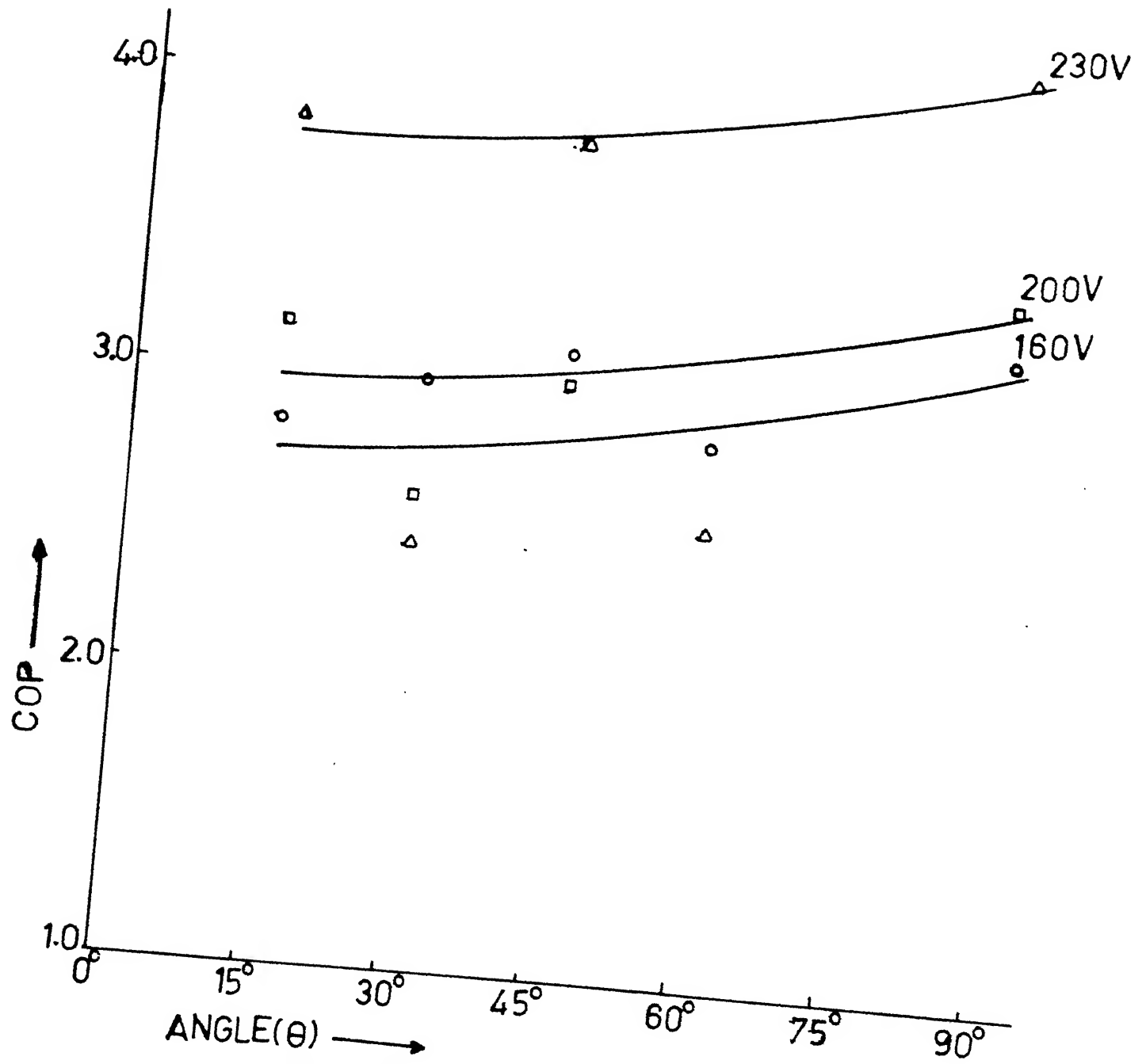
Fig. 3.20

COP vs Inclination angle for FD fan with double stage WDE
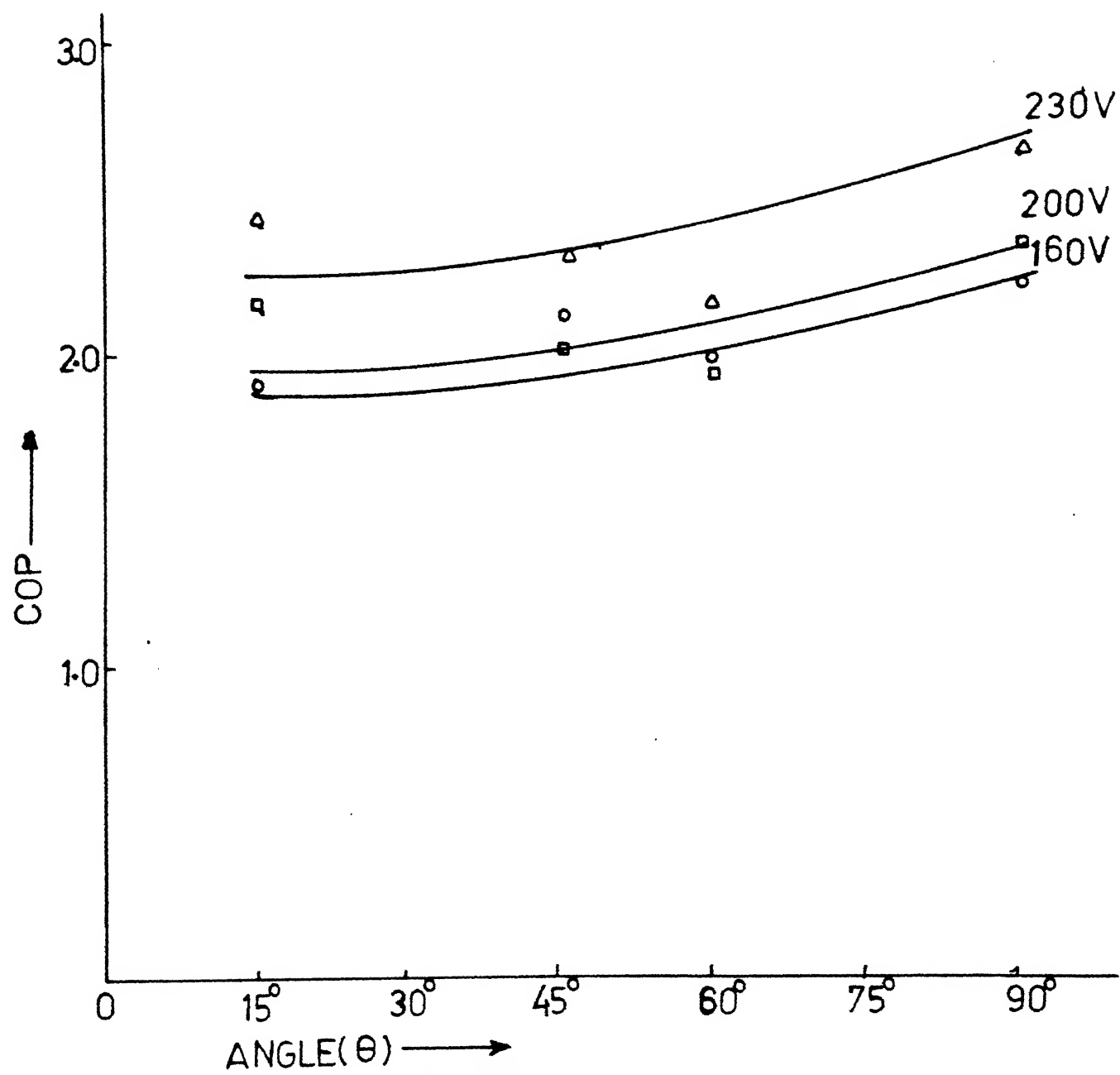
Fig. 3.21

COP vs Inclination angle for FD fan with single stage CDE

Fig. 3.22

COP vs Inclination angle for FD fan with double stage CDE

rate of the fan. Also when the number of stages increases, the net effect is a drop in COP due to increased resistance to flow and resulting lower air discharge rate. Comparative study of Figures 3.15 through 3.18 shows that COP of the system is more with FD fan than with ID fan for various cases. This can be explained on the basis at greater degree of turbulence is created by FD fan in the evaporative condenser box as compared with ID fan and this leads to higher rate of heat transfer from the condenser coils.

## 3.4    OPTIMUM ANGLE OF ORIENTATION:

For determination of an optimum angle of orientation for drift eliminators, running cost analysis was carried out. The main factors considered were power loss due to pressure drop across the drift eliminators and the amount of water lost to the atmosphere in the form of drift. The cost of water (for industrial purposes) was taken Rs. 1.25 per 1000 litres and the cost of power per kWhr was taken Rs. 0.70 for the purpose of cost analysis.

Initially, drift loss was calculated in terms of kg per hour and then multiplied by the cost per litre of water. This gave the cost of drift loss $(C_1)$ in terms of 'paise'.

Similarly, the power loss due to pressure drop across the drift eliminators was found out in terms of 'paise' per hour. The method of calculating power loss due to pressure drop is shown below.

Power loss due to pressure drop

$$= \Delta p \text{ x Volumetric discharge per hour } (V_{vol})$$

$$= (\rho_w gh) \text{ x } V_{vol} \qquad (3.1)$$

Power lost in terms of cost, $C_2$ (paise)

$$= \frac{(\rho_w gh \text{ x } V_{vol}) \text{ x } 70 \text{ x } 10^{-3}}{3.6 \text{ x } 10^6} \qquad (3.2)$$

where,

$\rho_w$ = density of water, $kg/m^3$

$g$ = acceleration due to gravity (9.80, $m/s^2$)

$h$ = pressure drop in mm of water

$V_{vol}$ = Volumetric discharge of air, $m^3/h$

Equation (3.2) gives the cost of power lost to overcome the pressure drop across the drift eliminators. The values of

Table 3.17

Drift Loss and Pressure Drop Costs for FD fan with
single stage CDE

Supply Voltage = 230 V AC
Entering Air velocity = 11.5 m/s

| Inclination Angle θ (degree) | Specific Drift Loss $m_d$ (kgw/kgda) | Pressure drop $\Delta p$ (mm of $H_2O$) | Drift loss $M_d$ (kg/hr) | $C_1$ (costs of water lost) (paise) | $C_2$ (Costs of pressure drop) (paise) |
|---|---|---|---|---|---|
| 15 | $0.4 \times 10^{-3}$ | 1.27 | 1.892 | 0.2365 | 1.019 |
| 45 | $1.0 \times 10^{-3}$ | 1.143 | 4.732 | 0.5915 | 0.917 |
| 60 | $1.1 \times 10^{-3}$ | 0.508 | 5.205 | 0.650 | 0.4077 |
| 90 | $2.2 \times 10^{-3}$ | 0.127 | 10.41 | 1.30 | 0.1019 |

## Table 3.18

### Drift Loss and Pressure drop Costs for ID fan with single stage CDE

Supply Voltage = 230 V (AC)
Entering air velocity = 7.5 m/s

| Inclination Angle $\theta$ (degree) | Specific Drift loss $m_d$ (kgw/kgda) | Pressure drop $\Delta p$ (mm of $H_2O$) | Drift loss $M_d$ (kg/hr) | $C_1$ (costs of water lost) (paise) | $C_2$ (Costs of pressure drop) (paise) |
|---|---|---|---|---|---|
| 15 | $0.4 \times 10^{-3}$ | 1.143 | 1.236 | 0.1545 | 0.5978 |
| 45 | $1.0 \times 10^{-3}$ | 1.016 | 3.091 | 0.3863 | 0.5314 |
| 60 | $1.6 \times 10^{-3}$ | 0.889 | 4.945 | 0.6181 | 0.4650 |
| 90 | $2.0 \times 10^{-3}$ | 0.127 | 6.182 | 0.7727 | 0.0664 |

Table 3.19

## Drift Loss and Pressure Drop Costs for ID fan with single stage WDE

Supply Voltage = 230 V AC
Entering Air Velocity = 7.66 m/s

| Angle $\theta$ (degree) | Specific Drift Loss $m_d$ (kgw/kgda) | Pressure drop $\Delta P$ (mm of $H_2O$) | Drift loss $M_d$ (kg/hr) | $C_1$ (Costs of water lost) (paise) | $C_2$ (Costs of pressure drop) (paise) |
|---|---|---|---|---|---|
| 15 | $1.0 \times 10^{-3}$ | 1.524 | 3.145 | 0.393 | 0.8147 |
| 30 | $1.2 \times 10^{-3}$ | 0.762 | 3.774 | 0.471 | 0.4073 |
| 45 | $1.9 \times 10^{-3}$ | 0.381 | 5.975 | 0.746 | 0.2036 |
| 60 | $2.0 \times 10^{-3}$ | 0.254 | 6.290 | 0.786 | 0.1357 |
| 90 | $2.6 \times 10^{-3}$ | 0.050 | 8.177 | 1.022 | 0.026 |

Table 3.20

Drift Loss and Pressure Drop costs for FD fan with double stage CDE

Supply Voltage = 230 V AC
Entering Air Velocity = 113.0 m/s

| Inclination Angle $\theta$ | Specific Drift loss $m_d$ | Pressure drop $\Delta P$ | Drift loss $M_d$ | $C_1$ (Costs of water lost) | $C_2$ (Costs of pressure drop) |
|---|---|---|---|---|---|
| (degree) | (kgw/kgda) | (mm of $H_2O$) | (kg/hr) | (paise) | (paise) |
| 15 | $0.3 \times 10^{-3}$ | 2.032 | 1.395 | 0.1743 | 1.6025 |
| 45 | $0.7 \times 10^{-3}$ | 1.270 | 3.255 | 0.4068 | 1.0015 |
| 60 | $1.0 \times 10^{-3}$ | 1.016 | 4.650 | 0.58125 | 0.7591 |
| 90 | $1.9 \times 10^{-3}$ | 0.127 | 8.835 | 1.1043 | 0.1001 |

Table 3.21

Drift Loss and Pressure Drop Costs for ID fan with double stages CDE

Supply Voltage = 230 V AC
Entering Air Velocity = 7.46 m/s

| Inclination Angle $\theta$ (degree) | Specific Drift loss $m_d$ (kgw/kgda) | Pressure drop $\Delta P$ (mm of $H_2O$) | Drift loss $M_d$ (kg/hr) | $C_1$ (Costs of water lost (paise) | $C_2$ (Costs of pressure drop (paise) |
|---|---|---|---|---|---|
| 15 | $0.4 \times 10^{-3}$ | 1.524 | 1.230 | 0.1537 | 0.893 |
| 45 | $0.6 \times 10^{-3}$ | 1.016 | 1.846 | 0.2307 | 0.5289 |
| 60 | $1.4 \times 10^{-3}$ | 0.889 | 4.307 | 0.5383 | 0.3628 |
| 90 | $1.8 \times 10^{-3}$ | 0.127 | 5.538 | 0.692 | 0.0661 |

## Table 3.22

Drift Loss and Pressure Drop Costs for ID fan with double stages WDE

Supply Voltage    = 230 V AC

Entering Air Velocity = 7.45 m/s

| Inclination Angle θ (degree) | Specific Drift loss $m_d$ (kgw/kgda) | Pressure drop $\Delta P$ (mm of $H_2O$) | Drift loss $M_d$ (kg/hr) | $C_1$ (Costs of water lost) (Paise) | $C_2$ (Costs of pressure drop) (paise) |
|---|---|---|---|---|---|
| 15 | $1.2 \times 10^{-3}$ | 2.032 | 3.670 | 0.458 | 1.056 |
| 30 | $1.4 \times 10^{-3}$ | 1.016 | 4.282 | 0.535 | 0.528 |
| 45 | $1.9 \times 10^{-3}$ | 0.888 | 5.811 | 0.7263 | 0.4622 |
| 60 | $2.2 \times 10^{-3}$ | 0.381 | 6.729 | 0.8411 | 0.1980 |
| 90 | $2.4 \times 10^{-3}$ | 0.050 | 2.341 | 0.9176 | 0.025 |

Table 3.23

Drift Loss and Pressure drop Costs for FD fan with single stage WDE

Supply Voltage = 230 V AC
Entering Air Velocity = 11.66 m/s

| Inclination Angle $\theta$ (degree) | Specific Drift loss $m_d$ (kgw/kgda) | Pressure drop $\Delta P$ (mm of $H_2O$) | Drift loss $M_d$ (kg/hr) | $C_1$ (Costs of water lost) (paise) | $C_2$ (Costs of Pressure drop) (paise) |
|---|---|---|---|---|---|
| 15 | $1.0 \times 10^{-3}$ | 2.667 | 4.798 | 0.599 | 2.170 |
| 30 | $1.4 \times 10^{-3}$ | 2.54 | 6.7175 | 0.838 | 2.066 |
| 45 | $2.0 \times 10^{-3}$ | 1.18 | 9.596 | 1.187 | 0.826 |
| 60 | $2.2 \times 10^{-3}$ | 0.508 | 10.55 | 1.312 | 0.4133 |
| 90 | $2.6 \times 10^{-3}$ | 0.127 | 13.43 | 1.675 | 0.1033 |

Table 3.24

Drift Loss and Pressure drop Costs for FD fan with double stages WDE

Supply Voltage = 230 V AC
Entering Air velocity = 11.55 m/s

| Inclination Angle θ (degree) | Specific Drift loss $m_d$ (kgw/kgda) | Pressure drop $\Delta P$ (mm of $H_2O$) | Drift loss $M_d$ (kg/hr) | $C_1$ (Costs of water lost) (paise) | $C_2$ (Costs of pressure drop) (paise) |
|---|---|---|---|---|---|
| 15 | $0.4 \times 10^{-3}$ | 3.048 | 1.89 | 0.236 | 2.44 |
| 30 | $1.2 \times 10^{-3}$ | 2.794 | 5.678 | 0.7097 | 2.242 |
| 45 | $1.7 \times 10^{-3}$ | 1.27 | 8.045 | 1.005 | 1.019 |
| 60 | $1.9 \times 10^{-3}$ | 0.762 | 8.991 | 1.123 | 0.6115 |
| 90 | $2.4 \times 10^{-3}$ | 0.127 | 11.35 | 1.418 | 0.1019 |
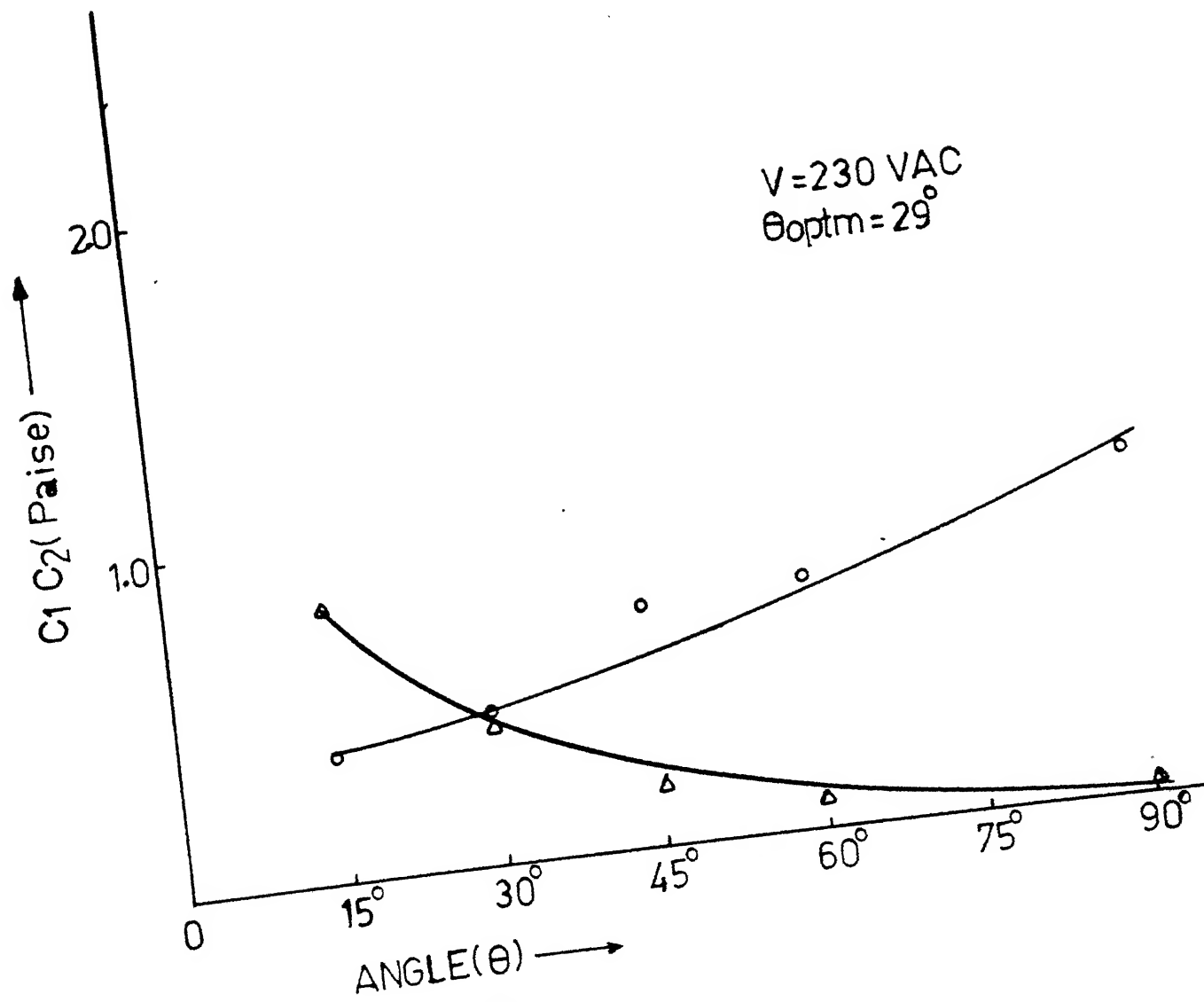
Fig. 3.23

$C_1, C_2$ vs Inclination angle for ID fan with single stage WDE
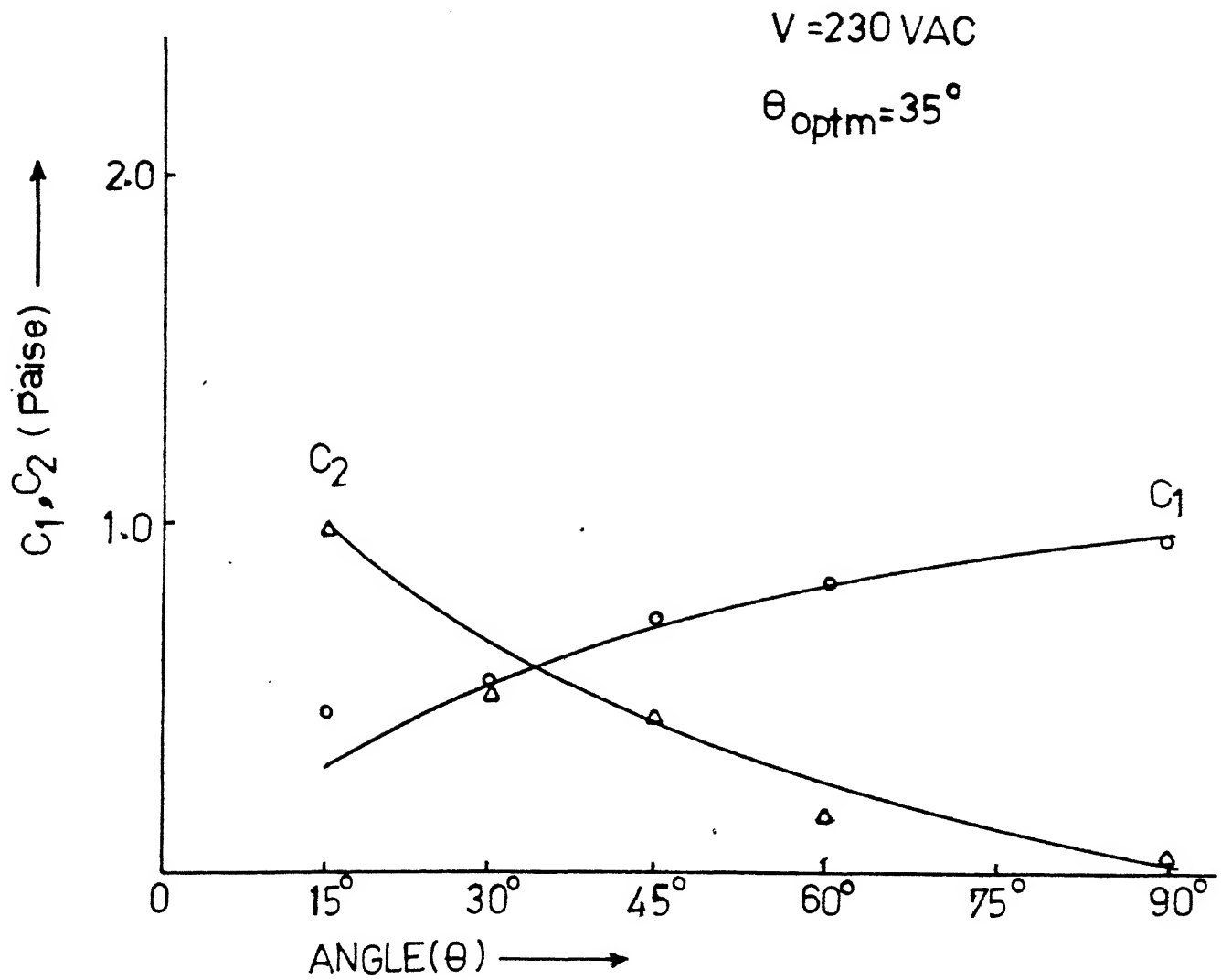
Fig. 3.24

$C_1, C_2$ vs Inclination angle for ID fan with double stage WDE
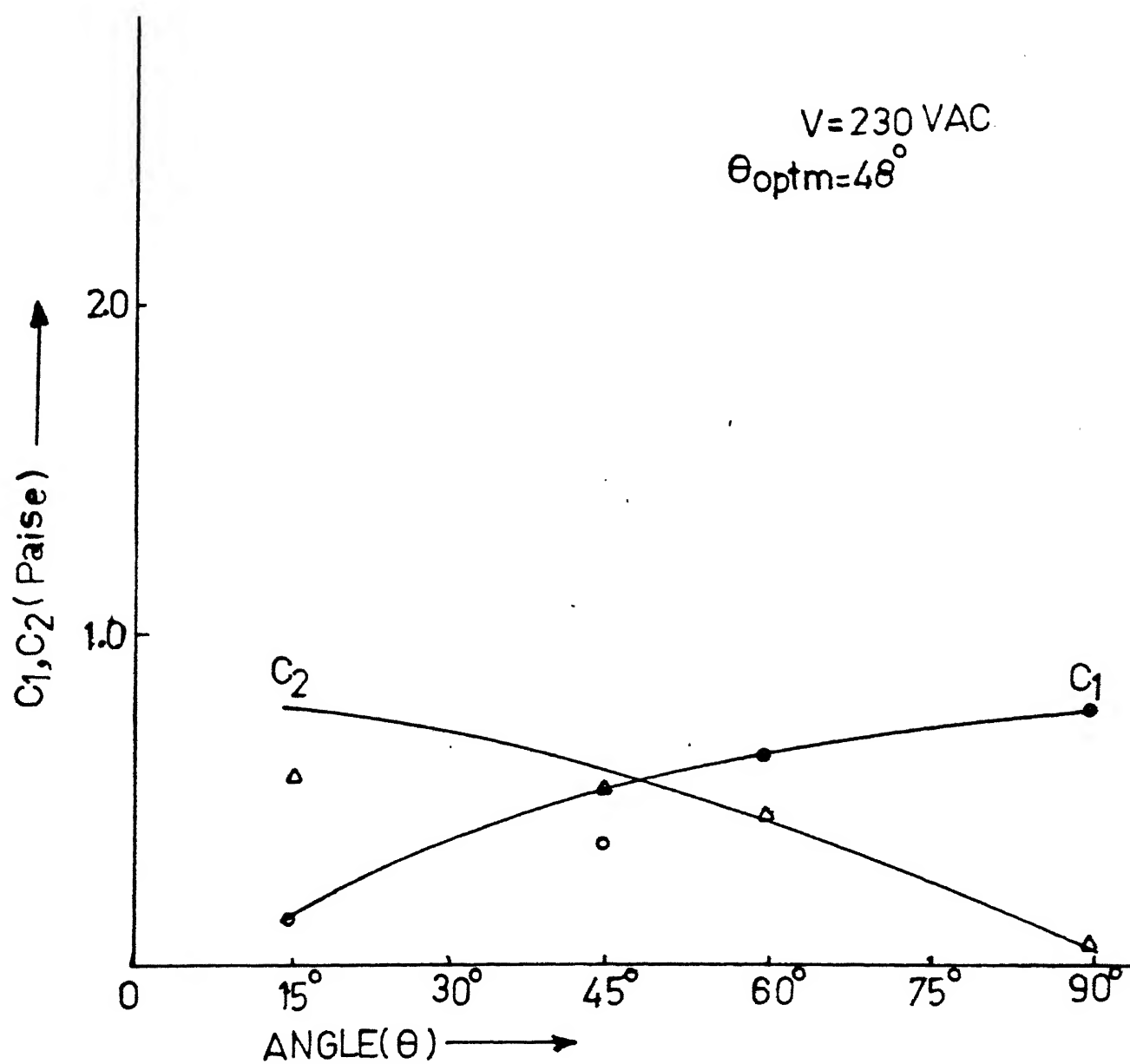
Fig. 3.25

$C_1, C_2$ vs Inclination angle for ID fan with single stage CDE

V=230 VAC

θoptm= 54°
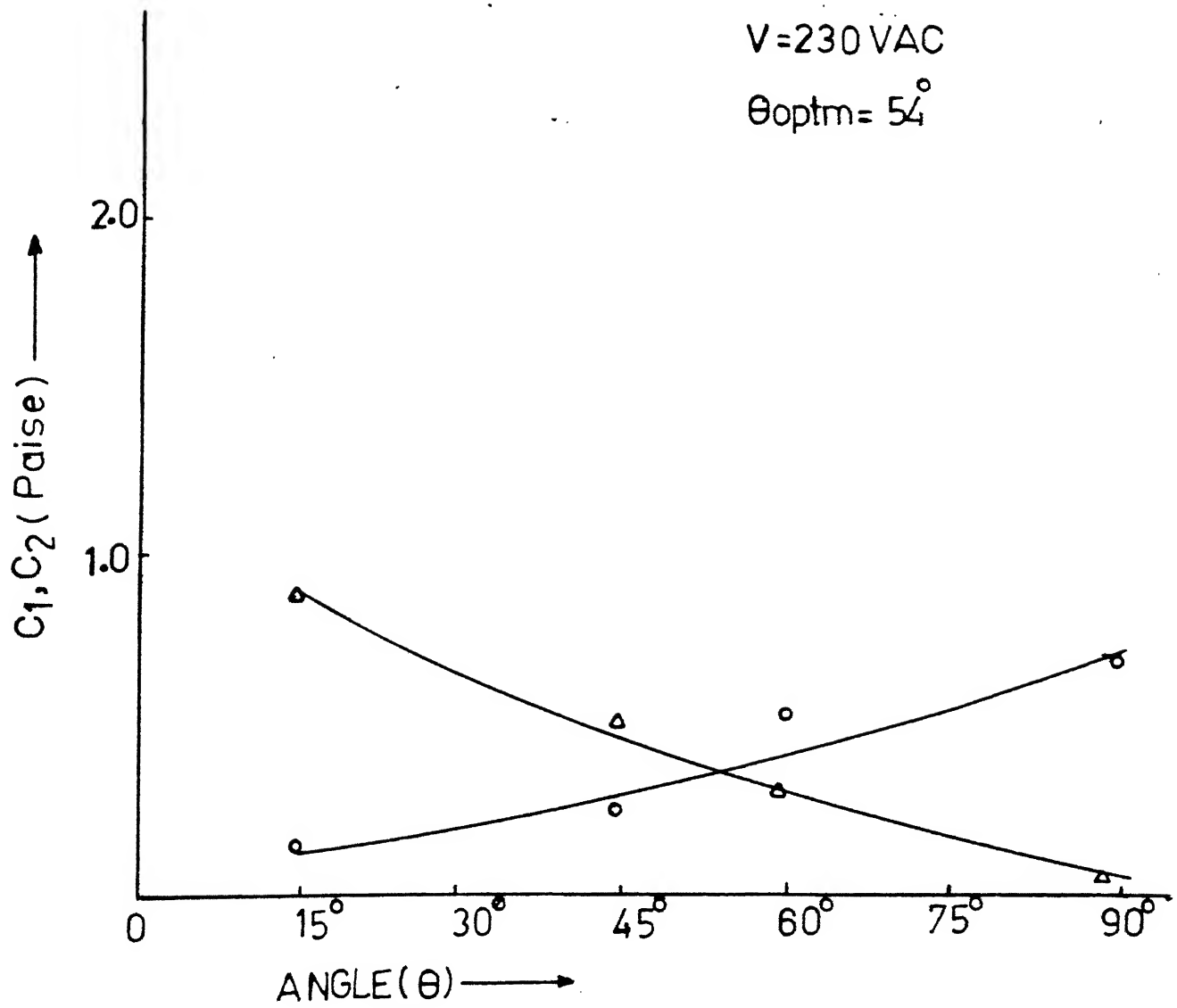
Fig. 3.26

$C_1, C_2$ vs Inclination angle for ID fan with double stage CDE
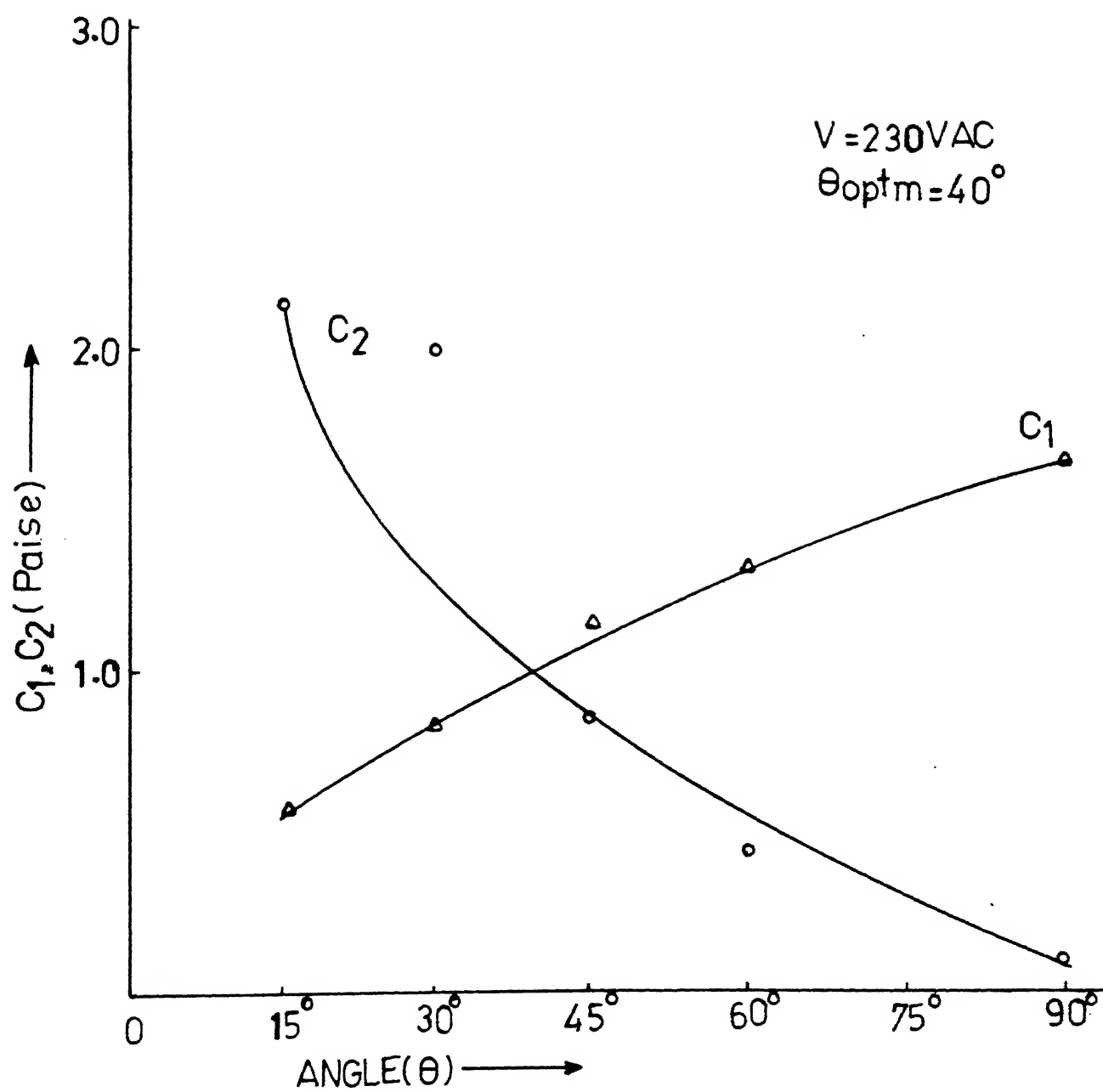
Fig. 3.27

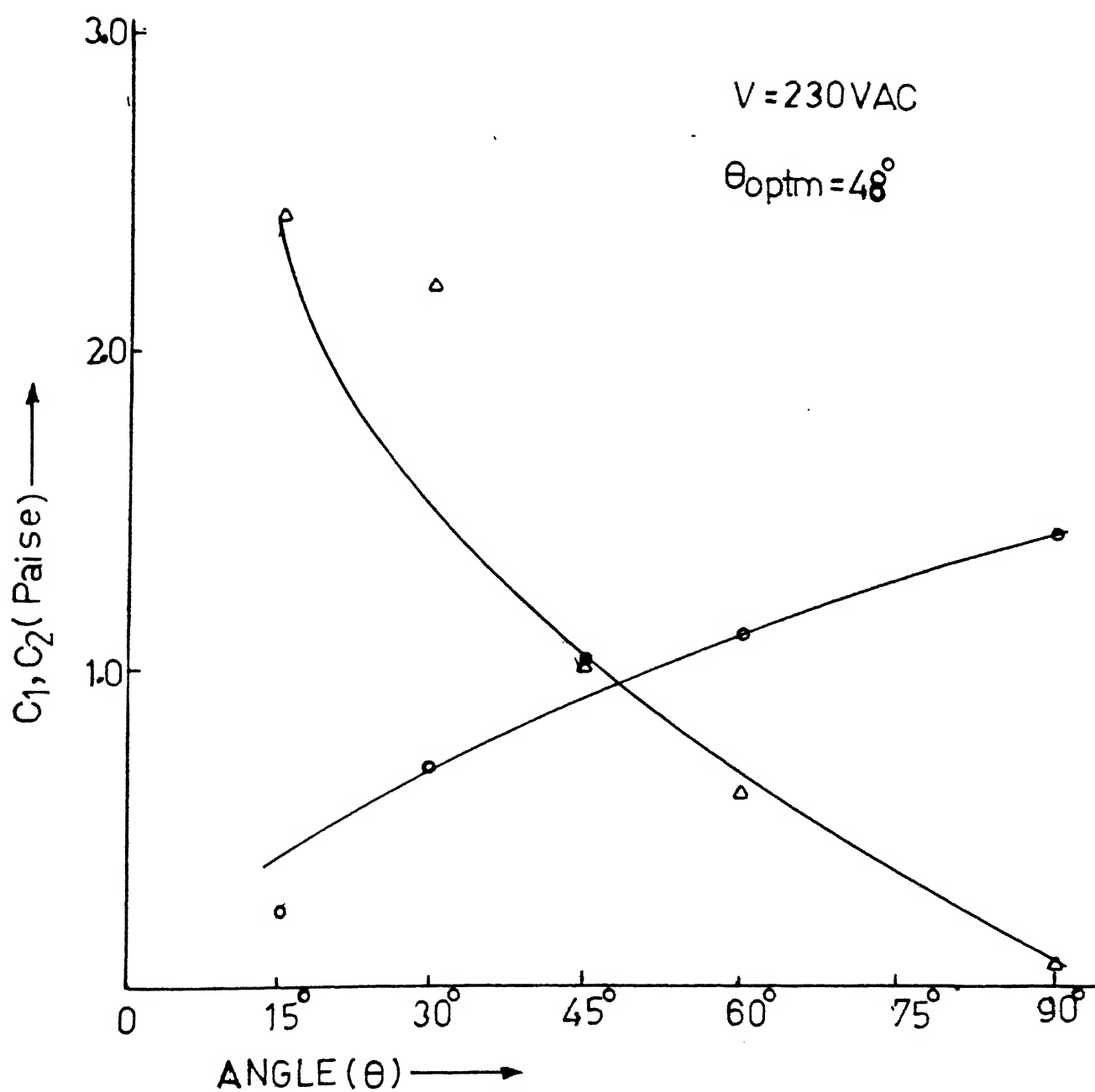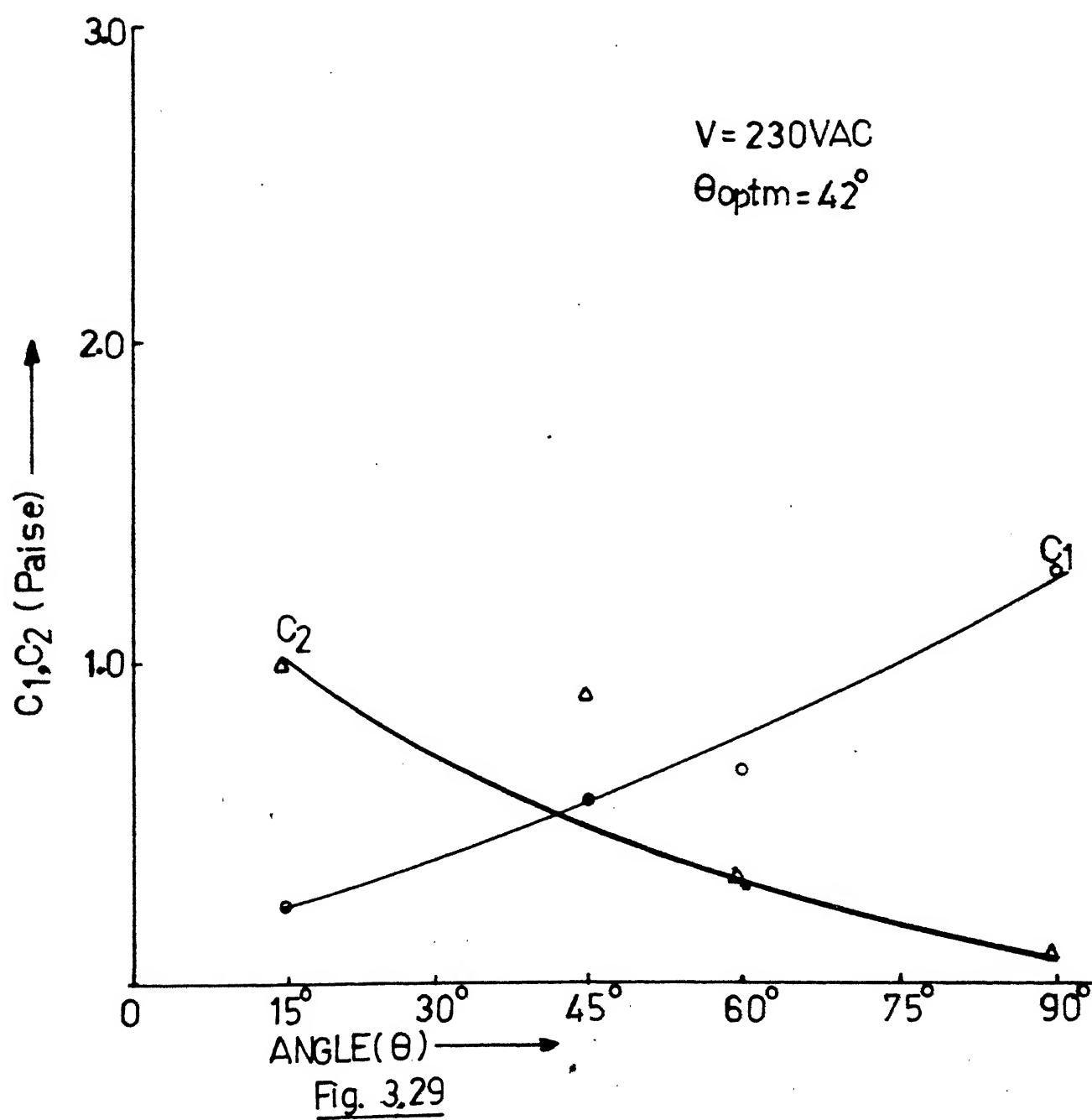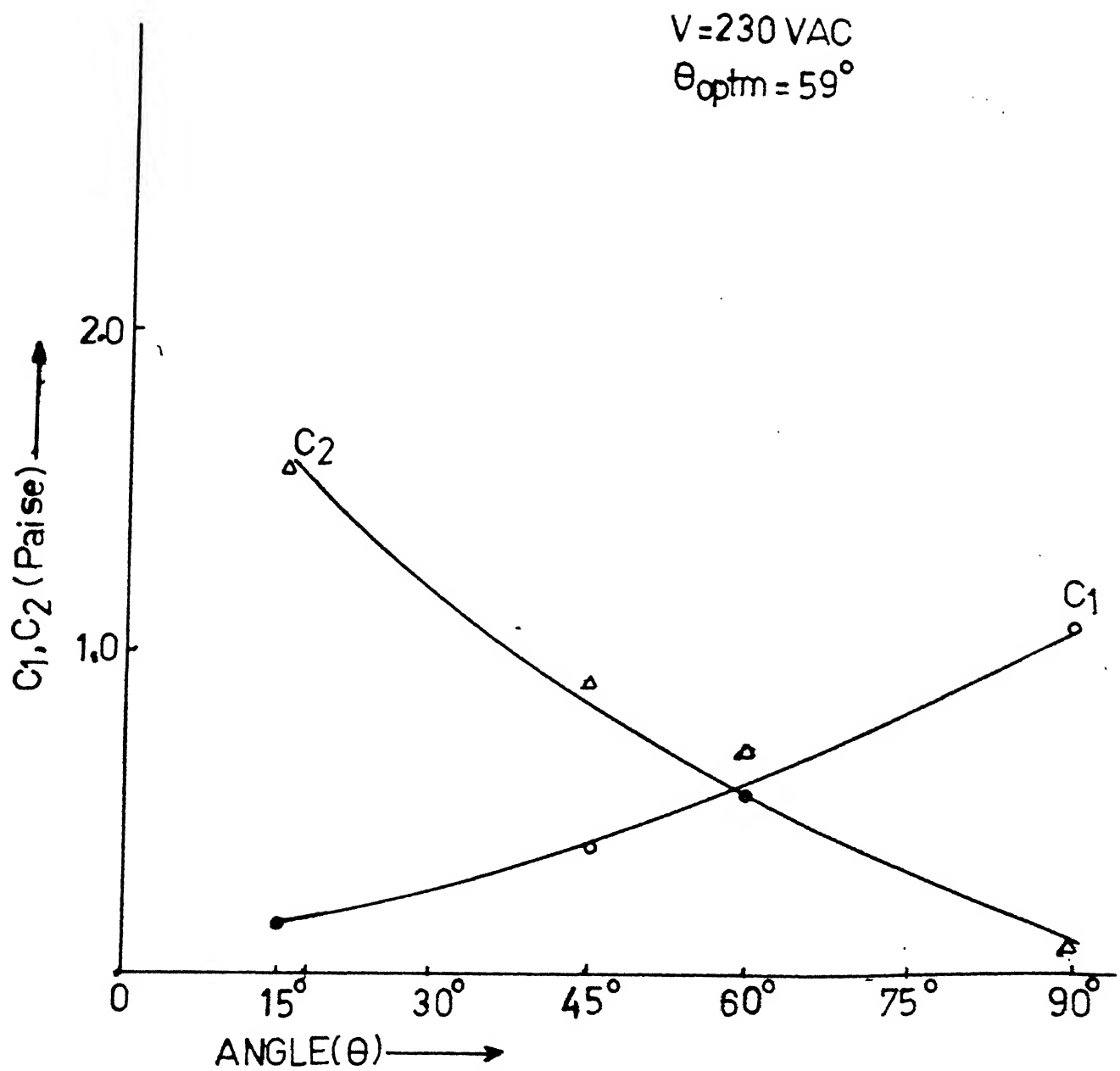$C_1, C_2$ vs inclination angle for FD fan with single stage WDE.

Fig. 3.28

$C_1, C_2$ vs Inclination angle for FD fan with double stage WDE

Fig. 3.29

$C_1, C_2$ vs Inclination angle for FD fan with single stage CDE

V = 230 VAC

$\theta_{optm} = 59°$

Fig. 3.30

$C_1, C_2$ vs Inclination angle for FD fan with double stage CDE

$C_1$, $C_2$ and $\dot{M}_d$ are given in Tables 3.18 to 3.24. The costs $C_1$ and $C_2$ are plotted versus $\theta$ (inclination angle) in Figures 3.23 to 3.30. It is found that for a particular value of $\theta$ for a given case, the values of $C_1$ and $C_2$ become equal, i.e., the cost of water lost to the atmoshphere per hour in the form of drift is same as the cost of power lost to overcome the pressure drop. This gives the optimum value of '$\theta$' (orientation angle) for drift eliminators for which the use of drift eliminator for a particular case becomes cost effective.

Similarly if the cost of pressure drop increases due to the increase in the capacity, cost of power or increased resistance due to algae deposits etc., the optimum value of '$\theta$' will be higher. It is also seen that for a given case, optimum value of '$\theta$' with single stage of drift eliminators will be different than that with multiple stages of drift eliminators. As the number of stages goes up, the optimum value of '$\theta$' also increases.

It can also be seen from these figures that for a given power tariff, if the drift increases for larger capacity of the unit, higher water circulation rate etc., the optimum $\theta$ will be lower.

3.5    CONCLUSIONS

1.    Drift loss and pressure drop are more with FD fan as compared to those with ID fan.

2.    Drift loss and pressure drop are less for concrete drift eliminators than those for wooden drift eliminators.

3.    Orientation of drift eliminators affects the performance of evaporative condenser. A higher inclination angle to the horizontal improves the COP of the system.

4.    As the number of drift eliminator stages increases, the drift loss decreases, but it leads to a higher pressure drop across the drift eliminator stages leading to an increase in the fan power input.

5.    An optimum angle of orientation for drift eliminators for single stage as well as multiple stages can be found out on the basis of cost analysis. Following optimum values of '$\theta$' were found out in the present investigation for different cases listed below.

(I)      $\theta = 29^{\circ}$      ID fan, single stage eliminators, supply voltage, 230 V AC, (Fig. 3.23).

(II)     $\theta = 36^{\circ}$      ID fan, double stage wooden drift eliminators, supply voltage, 230 V AC, (Fig. 3.24).

(III)    $\theta = 48^{\circ}$      ID fan, single stage concrete drift eliminators, supply voltage, 230 V AC, (Fig. 3.25).

(IV)     $\theta = 54^{\circ}$      ID fan, double stage concrete drift eliminators, supply voltage 230 V AC, (Fig. 3.26).

(V)      $\theta = 40^{\circ}$      FD fan, single stage wooden drift eliminators, supply voltage 230 V AC, (Fig. 3.27).

(VI)     $\theta = 48^{\circ}$      FD fan, double stage wooden drift eliminators, supply voltage 230 V AC, (Fig. 3.28).

(VII)    $\theta = 42^{\circ}$      FD fan, single stage concrete drift eliminators, supply voltage, 230 V AC, (Fig. 3.29).

(VIII)   $\theta = 59^{\circ}$      FD fan, double stage concrete drift eliminators, supply voltage, 230 V AC, (Fig. 3.30).

In industries, normally the angle of inclination used for drift eliminators is 45° which is close to angles found with the single stages used for different cases in the present experimental work. Generally it will suffice to use only single stage of drift eliminators, but due to safety considerations double stages are employed in big units.

3.6 SUGGESTIONS FOR FUTURE WORK:

The following additional research work can be takenup on the present test rig.

1. The test rig can be used for the testing of evaporative condensers.

2. The set up can be used for conducting experiments with drift eliminators of other geometrical shapes such as conical, corrugated and perforated, and data can be used for determining corelating factors for these geometrical shapes of drift eliminators.

3. By using different types of nozzles, one can also study the effect of mode of spray on system performance.

4. The set up can be used to determine the optimum values of fan speed and water circulation rate for a given capacity of refrigeration system using evaporative condenser.

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# APPENDIX

Inlet duct cross-sectional area = $0.1017 \, m^2$.

| Voltage V | Air flow rate $m^3$/min FD fan | Air flow rate, $m^3$/min ID fan |
|---|---|---|
| 160 | 45.520 | 37.832 |
| 180 | 46.751 | 38.625 |
| 200 | 52.843 | 40.639 |
| 220 | 56.931 | 42.714 |
| 230 | 61.020 | 45.520 |